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SACRAMENTO PLANT

INVESTIGATION OF SYSTEM COMPONENTS FOR HIGH CHAMBER PRESSURE ENGINES

Final Report

Contract NAS 8-5329

Part 1: Program Results

Report 5329-F

May 1964

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AEROJET-GENERAL CORPORATION

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INVESTIGATION OF SYSTEM COMPONENTS FOR HIGH CHAMBER PRESSURE ENGINES

Final Report

Part 1: Program Results

This report is in two parts:
Part 1: Program Results
Part 2: Appendixes

Prepared under

Contract NAS 8-5329

Prepared for

CHIEF, LIQUID PROPULSION TECHNOLOGY, CODE RPL
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Under Technical Direction of

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National Aeronautics and Space Administration
Huntsville, Alabama

AEROJET-GENERAL CORPORATION
A SUBSIDIARY OF THE GENERAL TIRE & RUBBER COMPANY

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FOREWORD

This final report presents a summary of the technical results produced under Contract NAS 8-5329, "Investigation of System Components for High Chamber Pressure Engines."

The contract was initiated by H. Burlage, Jr., Chief, Liquid Propulsion Technology (Code RPL), NASA Headquarters, Washington, D.C. The Headquarters Project Manager was F. E. Compitello (Code RPL); H. W. Fuhrmann, Chief, Propulsion Engineering Branch (Code R-P&VE-PM), Marshall Space Flight Center, Huntsville, Alabama was the program technical manager.

Work was conducted by the Research and Advanced Technology Division at the Liquid Rocket Plant, Sacramento, California, for the contract period of performance from 1 May 1963 to 31 March 1964, and was under the direction of R. Beichel, Division Manager, V. H. Ransom, Program Manager; and T. M. Weathers, Project Engineer.

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I. ABSTRACT

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This report describes the analyses performed to establish the fundamental design requirements for pressure- and size-affected auxiliary engine components such as valves, lines, bellows joints, and heat exchangers for high-thrust (1.5 to 24 x 10^6 lb) launch-vehicle engines. Problem areas and required technology associated with the design of these auxiliary components for five LO_2/LH_2 engines and one LO_2/RP-1 engine are shown, with most analysis performed on a 6 x 10^6 lb, LO_2/LH_2 gas-liquid staged-combustion cycle, single-pump, eight-chamber engine. Pump discharge pressures of up to 6000 psi are considered in order to establish those areas where advanced technology work should be performed on components, other than combustion chambers and pumps, necessary to operate liquid rocket engines with chamber pressures in the 2500- to 3500-psia range. The results of recent advanced-engine studies are combined with analytical procedures developed in the course of such engine programs as Titan I and II and M-1.

The study includes engine steady-state and transient analysis, off-design steady-state analysis, and the requirements of associated systems such as thrust vector control, thrust control, thrust modulation, propellant utilization, and tank pressurization. author

Several important problem areas, such as connectors and static and dynamic seals were not studied in detail because they are presently the subject of other studies; however, requirements and applications for these items are presented.

It is concluded that the design of the required auxiliary components for these engines is possible with present technology; however, advanced work in the fields of valve sealing, bellows joints, and filament-wound lines is recommended to ensure the design of efficient high-pressure components.

II. SUMMARY

Analyses have been performed to establish the fundamental design requirements for auxiliary engine components such as valves, lines, bellows joints, and heat exchangers for high-thrust (1.5 to 24 x 10^6 lb) launch-vehicle engines. Problem areas and required technology associated with the design of pressure-affected auxiliary components have been determined for five LO_2/LH_2 engines and one LO_2/RP-1 engine, including various combinations of engine cycle (gas generator, gas-liquid staged combustion, and gas-gas staged combustion) and engine configuration (single engine, multiple chamber--single pump, and modular engine). These engines were designed and analyzed using available advanced-engine concepts and existing analytical techniques established in the course of such engine programs as Titan I and II and M-1.

Nominal pump discharge pressures up to 6000 psi were considered in order to evaluate the component requirements of engines with chamber pressures in the 2500- to 3500-psi range where greater overall engine efficiency may be realized, as shown by various engine studies.*

The study covers engine-design procedures, steady-state as well as start and shutdown transient analysis, off-design steady-state analysis, and the requirements of associated systems such as thrust vector control, thrust control, thrust modulation, propellant utilization, and tank pressurization.

The auxiliary components analyzed during this program were limited to those whose design is a function of both the high pressure and high thrust level of an advanced launch-vehicle liquid rocket engine. The only exception is some evaluation of suction lines and valves in order to establish the relative severity of large-size problems as compared with those of large size and high pressure combined. Because of this specialized interest, most of the small-sized hardware, and those

*Design Study of Large Unconventional Liquid Propellant Rocket Engines and Vehicles, Aerojet-General Report R-LRP-257, 18 April 1962 (C).

Study of large Unconventional Liquid Propellant Rocket Engines and Vehicles, North American Aviation Report R-3517D, 30 April 1962 (C).

High Chamber Pressure Operation for Launch Vehicle Engines, Aerojet-General Report 4008-F-1, 15 April 1963 (U).

Design Study of a High Thrust, High Pressure, Gas-Gas Staged Combustion Cycle LOX/LH_2 Engine, Aerojet-General Report 9400-1S, May 1964.

II, Summary (cont.)

components not experiencing high pressure in the engine system, are not discussed in this report. It is considered that these other components, while requiring new designs in many instances, are within the state-of-the-art and may be produced without requiring specific technological advances. The purpose of this effort, rather, is to establish what, if any, advanced technology effort should accompany that presently in progress on high-pressure combustion and high-pressure pumping systems so that no auxiliary component requirement would delay the development of an advanced high-pressure launch vehicle engine system.

Component sizes and pressures were established for various functional requirements. Discussions are included on the applicability of (1) various types of valves such as ball, butterfly, sleeve, poppet, and venturi and (2) unique valves which are integrated with the propellant pumps. This valve study consisted of conceptual or preliminary design to establish sizes and pressure drops, followed by parametric analysis. No attempt was made to solve the very important problems of valve sealing, because these are the subject of other efforts (Appendix A), nor were any detailed valve designs prepared. No experimental work was performed in this program.

Pressure drop and structural considerations of propellant and hot-gas lines have been evaluated, and the results are presented. The use of available piping-system-analysis computer services is recommended for large high-pressure engines. Required technology has been determined for bellows-type flexible joints: it is shown to require up to 20-in.-dia bellows with proof pressures from 8000 to 13,000 psia. The feasibility of using filament-wound reinforced plastics for the fabrication of lines and ducts is shown as a potential means for obtaining limited line-flexibility without the use of bellows joints. No detailed study of connector and static seal problems was made during this program, although it is anticipated that other, more specialized programs will utilize the size, pressure, temperature, and flow parameters resulting from this effort in the comprehensive study of connector and seal problems. One such program is Contract NAS 8-11523, "Zero Leakage Design for Duct and Tube Connections for Deep Space Travel" presently being conducted by General Electric Advanced Technology Laboratories, Schenectady, New York.

II, Summary (cont.)

Table II-1 summarizes the critical parameters of the representative lines analyzed and delineates the functional requirements for valves for each of these lines. The potential applicability of the various valve types is shown in each case. The probability that bellows will be required for each line is shown; this probability is based on an evaluation of the line configuration, inherent line flexibility, pressure, and estimated thermal gradients in the engine, and not on a detailed structural analysis of each line application. Similarly, the applicability of various valve types is based primarily on evaluation of the line diameter and configuration, pressure, allowable pressure drop, and envelope restrictions, rather than on consideration of the feasibility or cost of actually developing a particular valve to fit the requirements. For example, it will be noted that sleeve and poppet valves are rated the same in most applications, although the present state of sealing technology may be expected to make the poppet valve more feasible to develop for these conditions.

The results of a parametric heat-exchanger study and the preliminary design of a sample heat-exchanger are presented in this report. Table II-2 provides a summary of hot-gas and propellant parameters that constitute the essential heat-exchanger design criteria for each of the engines investigated. As with the other components, no detailed design work was performed.

While it is concluded that the design of the required auxiliary components for these engines is possible with present technology, the advanced-technology work recommended in the areas of valve sealing, bellows joints, and filament-wound lines will permit the design of efficient hardware.

TABLE II-1

DESIGN CRITERIA SUMMARY, HIGH PRESSURE FLUID TRANSFER

Engine Description / Most Critical Lines

Designation	F (LB)	Configuration	Propellants	Cycle	P_o Fuel (PSI)	Number of Chambers	Ref. ΔP Calc. Fig. No.	D Dia. (In.)	Fluid	$\dot{w}$ (LB/SEC)	P (PSI)	T (°F)	ΔP Allowed (PSI)	ΔP Calculated (PSI)	U Anchor to Anchor Distance (In.)	L (In.)	L/D	Inherent Flexibility Factor $\frac{D}{(L-U)^2}$	Probability That Bellows Will Be Required
AJ-1	6M̄	MCSP	LO_2/LH_2	STAGED COMBUSTION	4550	8	VIII-B-1	6.5	LH_2	278	4100	-210	25	22.4	58	64	9.855	0.1805	HIGH
							-2	5.0	LH_2	278	4550	-423	25	21.9	46	60	12.0	.0255	FAIR
							-4	4.0	LO_2	251	4100	-293	25	15.2	32	40	10.0	.0625	FAIR
							-	6.5	LO_2	1680	4100	-293	50	N	30	38	8.95	.1016	HIGH
							-	7.0	HOT GAS	628	2734	1400	N	N	N	N	N	N	N
AJ-1 (2000)	6M̄	MCSP	LO_2/LH_2	STAGED COMBUSTION	2000	8	-1	6.5	LH_2	278	1680	-210	18	N	58	64	9.855	.1805	HIGH
							-2	5.0	LH_2	278	2000	-423	17	N	46	60	12	.0255	FAIR
							-4	4.0	LO_2	251	1740	-293	20	N	32	40	10	.0625	FAIR
							-	6.5	LO_2	1680	1740	-293	20	N	30	38	8.45	.1016	HIGH
							-	7.0	HOT GAS	628	1362	1400	N	N	N	N	N	N	N
AJ-1 (6000)	6M̄	MCSP	LO_2/LH_2	STAGED COMBUSTION	6000	8	-1	6.5	LH_2	278	5375	-210	25	N	58	64	9.855	.1805	HIGH
							-2	5.0	LH_2	278	6000	-423	25	N	46	60	12	.0255	FAIR
							-4	4.0	LO_2	251	5275	-293	25	N	32	40	10	.0625	FAIR
							-	6.5	LO_2	1680	5275	-293	50	N	30	38	8.45	.1016	HIGH
							-	7.0	HOT GAS	628	3350	1400	N	N	N	N	N	N	N
AJ-2A	24M̄	MCSP	LO_2/LH_2	STAGED COMBUSTION	4679	12	-6	8.0	LH_2	740	4679	-403	25	22.0	51	54	6.755	.889	HIGH
							-7	8.0	LH_2	740	4088	-413	25	17.4	71	78	9.75	1.142	HIGH
							-8	8.0	LO_2	3825	4101	-290	50	38.4	43	47	5.88	.5	FAIR
							-9	8.0	LH_2	740	4088	-403	25	14.7	28	34	4.255	.2221	HIGH
AJ-3	6M	MCSP	LO_2/RP1	STAGED COMBUSTION	4000	8	-11	1.5	RP 1	47	3900	100	50	22.4	32	35	23.35	.1668	FAIR
							-12	4.0	RP 1	640	4000	80	50	41.5	51	59	14.75	.0625	FAIR
AJ-5	6M̄	SCSP	LO_2/LH_2	GAS GENERATOR	4100	1	-13	6.75	LH_2	742	4075	-410	50	48.0	67	71	10.52	.422	VERY HIGH
							-14	7.5	LH_2	1446	4075	-410	370	298	25	28	3.735	.833	VERY HIGH
							-15	17.0	LO_2	13,127	4075	-290	50	45.0	62	139	8.17	.02862	LOW
							-16	17.9	HOT GAS	512	540	690	15	10.5	106	136	7.6	.01989	HIGH
							-17	12.7	HOT GAS	256	540	690	15	12.3	106	136	10.71	.0141	FAIR
AJ-6	6M	SCSP	LO_2/LH_2	STAGED COMBUSTION	4550	1	-18	10.0	LH_2	1119	3960	-210	50	28.8	90	91	9.1	10.0	VERY HIGH
							-19	4.75	LO_2	1024	4000	-290	50	50.0	49	59	12.42	.0475	LOW
AJ-8A	1.5M	SCSP MODULE	LO_2/LH_2	GAS/GAS STAGED COMBUSTION	4930	1	VI-A-18 (ENG.)	8.5	LH_2	578	4930	-410	30	-	43	60	7.055	.02945	LOW
							VI-A-18	7.5	LO_2	3466	4930	-290	65	-	37	53	7.06	.02935	LOW
AJ-8B	1.5M̄	SCSP MODULE	LO_2/LH_2	GAS/GAS STAGED COMBUSTION	4930	1	VIII-820	12.8	HOT GAS (O_2 RICH)	1567	3000	1062	160	20.0	48	99	7.74	.00492	LOW
							-21&-22	12.3	HOT GAS	380	3000	1059	160	20.0	91	103	8.375	.0854	LOW

Valve Requirements and Application

(Functional Requirement Probability 1-Low 2-Fair 3-High; Applicable Valve Types 1-Poor 2-Fair 3-Good)

Designation	Ref. ΔP Calc. Fig. No.	On/Off: Start/Shutdown	Modulating: Thrust Modulation	Modulating: Thrust Control	PU	TVC	ΔP Allowable (PSI)	Sleeve Inline	Sleeve Angle	Poppet Inline	Poppet Angle	Rotary Sleeve (Angle)	Butterfly	Ball Inline	Ball Angle	Venturi Poppet Inline	Venturi Poppet Angle
AJ-1	VIII-B-1	1	1	1	1	NA	50	3	2	3	2	1	1	2	1	3	2
	-2	3	3	3	1	NA	50	2	1	2	1	1	1	3	-	3	1
	-4	3	1	3	1	NA	50	3	3	3	3	1	1	3	1	3	2
	-	3	3	3	2	NA	100	-	3	-	3	1	1	1	2	2	2
	-	NA	NA	NA	NA	3	-	3	3	3	3	2	1	1	1	3	2
AJ-1 (2000)	-1	1	1	1	1	NA	35	3	2	3	2	1	2	2	1	3	2
	-2	3	3	3	1	NA	35	2	1	2	1	1	2	3	-	3	1
	-4	3	1	3	1	NA	50	3	3	3	3	1	2	3	1	3	2
	-	3	3	3	2	NA	75	-	3	-	3	1	2	1	2	2	2
	-	NA	NA	NA	NA	3	-	3	3	3	3	2	1	1	1	3	2
AJ-1 (6000)	-1	1	1	1	1	NA	50	3	2	3	2	1	1	1	1	3	2
	-2	3	3	3	1	NA	50	2	1	2	1	1	1	2	-	3	1
	-4	3	1	3	1	NA	50	3	3	3	3	1	1	3	1	3	2
	-	3	3	3	2	NA	150	-	3	-	3	1	1	1	1	2	2
	-	NA	NA	NA	NA	3	-	3	3	3	3	2	1	1	1	3	2
AJ-2A	-6	3	3	3	1	NA	49	3	3	3	3	1	1	1	1	3	1
	-7	1	1	1	1	NA	49	3	2	3	2	1	1	1	1	3	2
	-8	3	3	3	2	NA	51	3	3	3	3	1	1	1	1	2	3
	-9	1	1	1	1	NA	49	-	3	-	3	1	1	1	1	2	1
AJ-3	-11	3	1	3	1	NA	50	3	3	3	3	2	1	3	2	3	-
	-12	3	3	3	2	NA	50	3	2	3	2	1	1	2	1	3	1
AJ-5	-13	3	1	1	1	NA	50	3	3	3	3	1	1	2	2	3	3
	-14	2	2	2	1	NA	50	1	3	1	3	1	1	2	2	1	3
	-15	2	2	1	1	NA	50	1	3	1	3	3	1	1	1	1	2
	-16	-	1	1	2	NA	MIN.	3	1	3	1	1	2	1	1	1	1
	-17	-	1	1	2	NA	MIN.	3	1	3	1	1	2	1	1	1	1
AJ-6	-18	3	3	3	1	NA	50	3	-	3	-	-	1	1	-	3	-
	-19	3	3	3	2	NA	100	3	2	3	2	1	1	3	1	3	1
AJ-8A	VI-A-18 (ENG.)	3	3	3	1	NA	120	2	1	2	1	1	1	2	1	3	1
	VI-A-18	3	3	3	1	NA	70	2	1	2	1	1	1	2	1	3	1
AJ-8B	VIII-820	NONE															
	-21&-22	NONE															

(1) N = NO ANALYSIS PERFORMED

(2) NA = NOT APPLICABLE

TABLE II-2

DESIGN CRITERIA SUMMARY, HEAT EXCHANGERS

ENGINE DESCRIPTION							HOT GAS					CRYOGENIC PROPELLANT					
								TURBINE INLET		TURBINE EXHAUST		LO_2		LH_2			
DESIGNATION	F (LB)	CONFIGURATION	PROPELLANTS	CYCLE	P_O FUEL (PSI)	NUMBER OF CHAMBERS	$\dot{w}$ (LB/SEC)	P (PSI)	T (°F)	P (PSI)	T (°F)	P (PSI)	T (°F)	P (COLD) (PSI)	T(COLD) (°F)	P (HOT)* (PSI)	T(HOT)* (°F)
AJ-1	6M	MCSP	LO_2/LH_2	STAGED COMBUSTION	4550	8	4285	3496	1440	2734	1344	4100	-293	4550	-423	4100	-210
AJ-1 (2000)	6M	MCSP	LO_2/LH_2	STAGED COMBUSTION	2000	8	4285	1520	1440	1362	1397	1740	-293	2000	-423	1680	-310
AJ-1 (6000)	6M	MCSP	LO_2/LH_2	STAGED COMBUSTION	6000	8	4285	4685	1440	3350	1314	5375	-293	6000	-423	5400	- 60
AJ-2A	24M	MCSP	LO_2/LH_2	STAGED COMBUSTION	4679	12	17,142	3620	1440	2730	1340	4101	-293	4679	-423	4088	-210
AJ-3	6M	MCSP	LO_2/RP1	STAGED COMBUSTION	4000	8	14,675	3590	1313	2700	1224	4000	-293	NOT APPLICABLE			
AJ-5	6M	SCSP	LO_2/LH_2	GAS GENERATOR	4100	1	545 (OXIDIZER) 545 (FUEL)	427 3570	909 .440	174 427	708 909	4100	-293	4100	-423	3630	-310
AJ-6	6M	SCSP	LO_2/LH_2	STAGED COMBUSTION	4550	1	4285	3496	1440	2734	1344	4100	-293	4550	-423	4100	-210
AJ-8A	1.5M	SCSP MODULE	LO_2/LH_2	GAS/GAS STAGED COMBUSTION	4930	1	3134 (OXIDIZER) 760 (FUEL)	4192 4192	1141 1160	3000 3000	1062 1059	4931	-293	4931	-423	3370	-310
AJ-8B	1.5M	SCSP MODULE	LO_2/LH_2	GAS/GAS STAGED COMBUSTION	4930	1	SAME AS AJ-8A										

* HOT VALUES FOR LH_2 ARE FOR FUEL WHICH HAS BEEN USED FOR REGENERATIVE COOLING.

III. CONCLUSIONS

A. There are no problems associated with the design of high-pressure auxiliary components that cannot be solved or circumvented by a "brute force" approach with present technology.

B. The design of auxiliary hardware, such as valves, lines or ducts, and bellows joints, which is efficient from the standpoint of functional performance and weight, will be greatly facilitated if the following work is first performed:

1. A program to incorporate the results of recent and current seal studies into the design of large high-pressure poppet, sleeve, ball, and venturi valves.

2. A program to design, fabricate, and test a large high-pressure hinged bellows joint with emphasis on the accuracy of predicting stability and spring rate at various pressures.

3. A program to demonstrate the feasibility of designing and fabricating subscale filament-wound reinforced-plastic propellant lines for high-pressure service.

C. Present technology is adequate for the design of efficient heat exchangers for large high-pressure rocket engines.

D. Comprehensive structural analysis of each high-pressure-line application is necessary to minimize and/or design for end-reaction loads of high-pressure, large-diameter metal or plastic lines and to establish the need for flexible joints and the specifications for such joints. Available computer programs for piping system analysis may be utilized for this task.

E. The combined effects of high pressure and high flow rate require that valve designs for pump-discharge and thrust-chamber valves incorporate concepts

III, Conclusions (cont.)

different from those used on most engines of current technology. No single valve concept can possibly meet all of the potential requirements of various engine applications. As done on all flight-weight thrust-chamber valves of engines such as Bomarc, Atlas, Titan, M-1, and Apollo, these large, high-pressure valves will probably be developed directly by the engine manufacturer to suit the specific engine requirements rather than be purchased from a valve manufacturer as are most vehicle-system valves. Pressure-balanced poppet valves, venturi poppet valves, and sleeve valves offer the most promise for these applications with pressure-unbalanced poppet valves, rotating sleeve valves, and ball valves having limited application. Until such time as sliding seal technology is advanced substantially, sealing problems will probably dictate the use of some type of poppet valve for large diameter, high-pressure applications.

F. The only up-to-date design information pertaining to large, high-pressure, multiple-ply-formed bellows for flexible joints is proprietary to the various bellows vendors and is not available in the literature. While confidence is expressed that such units can be designed, some doubts exist relative to the techniques for forming the bellows to meet the specified design. A demonstration program is required to verify the accuracy of the design techniques and to indicate the performance that may be expected from large bellows at high pressure.

G. Filament-wound reinforced-plastic lines offer the potential of up to 100 times the flexibility of metal lines of the same geometry. However, problems such as those associated with liners and connectors remain to be solved technically and economically.

IV. RECOMMENDATIONS

A. ADVANCED TECHNOLOGY

It is recommended that the three programs outlined in Sections VII-G, VIII-E, and VIII-H of this report be initiated. These programs include (1) application of the latest advances in seal technology to large high-pressure valves, (2) the demonstration of large, high-pressure bellows technology, and (3) demonstration of the feasibility of using filament-wound reinforced-plastic to obtain flexibility in high-pressure rocket-engine lines and ducts.

The three programs, while concerned with providing information that will permit the design of components now beyond the state of the art, are each of a somewhat different nature. The valve-sealing program is concerned with the application of newly-developed technology, the bellows-joint program will determine the success with which present technology can be extrapolated to appreciably more severe applications, and the filament-wound reinforced-plastic line program will demonstrate the feasibility of applying a relatively new technology to a totally new application.

The nature of the valve and bellows programs (primarily associated with size and pressure effects) will necessitate at least some full-scale testing to verify results. It is expected that suitable scaling factors can be established for the filament-wound line that will permit subscale testing to satisfy the feasibility-demonstration requirements.

The 20-in.-dia size and 6600-psi operating pressure recommended for the valve and bellows programs are based on the engines considered in this program. None of these limiting size and pressure requirements were established by the 24×10^6-lb thrust AJ-2 engine because of the multiple-chamber configuration. The 20-in./6600-psi requirement will cover any forseeable engine with a combustion chamber size up to 6×10^6-lb thrust, with the exception of certain gas/gas cycle configurations where extrapolation of AJ-8B line diameters (Figure II-1) shows a potential 26-in. requirement. If engines with nominal operating pressures below

IV, A, Advanced Technology (cont.)

6000 psi or thrust levels below 6×10^6-lb thrust are anticipated, the following rule of thumb may be used for establishing the size and pressure specifications for the recommended valve and bellows programs:

Engine Cycle / Component Design Parameter / Medium	Gas Generator or Staged Combustion		Gas/Gas Staged Combustion	
	Propellant	Hot Gas	Propellant	Hot Gas
Pressure (psi)	1.1 Pd	0.8 x 1.1 Pd	1.1 Pd	0.6 x 1.1 Pd
Diameter (in.)	$20\sqrt{\frac{F_c}{6}}$	$20\sqrt{\frac{F_c}{6}}$	$20\sqrt{\frac{F_c}{6}}$	$13\sqrt{\frac{F_c}{1.5}}$

P_d = Nominal Pump Discharge Pressure, psia

F_c = Nominal Engine Thrust Per Combustor, $\text{lb} \times 10^6$

$$\text{(i.e., for AJ-1, } F_c = \frac{6}{8}\ \frac{\text{(Engine thrust)}}{\text{(No. of Combustors)}} = 0.75\text{)}$$

It is further recommended that present investigations of static seals and connectors be continued to provide criteria for lines up to 20 in. in diameter and 6600 psi, or such size and pressure as determined to be appropriate by the rough rule of thumb described above.

B. APPLICATION OF PRESENT TECHNOLOGY

It is also recommended that the design of auxiliary components for large high-pressure engines include consideration of the items discussed in Section III, such as the use of existing computer programs for the analysis of piping systems and evaluation of valve-sealing requirements based on each particular valve function. This latter consideration would not require valve seals to seal under high pressure when the valve is normally open; yet it would require tight sealing during the closed, low-pressure condition. The use of poppet valves, possibly with bellows for secondary or internal seals, is recommended until such time as better sliding seals are available.

V. INTRODUCTION

A. PROGRAM OBJECTIVES

Studies of propulsion systems for the next generation of launch vehicles have indicated potential requirements for cryogenic liquid-rocket-engine systems designed for thrust levels up to 24×10^6 lb.* The optimum rocket-engine chamber pressures for these systems are from two- to four-times greater than those of currently operational engines. These higher chamber pressures, in turn, require propellant pumps with discharge pressures up to 6000 psia. The more obvious problems (turbopump design, nozzle heat transfer, combustion instability, nozzle performance, etc.) associated with development of primary components of such engine systems have been the subject of a variety of advanced-technology studies throughout the aerospace industry. The successful development of these engine systems will depend not only on the successful performance of primary components, but also on the auxiliary components (such as valves, ducts, heat exchangers, and bellows) necessary to integrate the primary components into an engine system. The purpose of this program was to evaluate the feasibility of developing the auxiliary components, the design of which may be critically influenced by large size and high pressure, and to identify the required areas of technology.

The program objectives required the following three primary accomplishments:

1. The establishment (by means of comprehensive design and analysis of the engine systems) of basic design criteria for the lines or ducts, bellows, joints, valves, and heat exchangers required for large high-pressure engines.

2. The determination of the problems and required technology associated with design of the auxiliary components.

3. The recommendation of specific advanced-technology efforts where the available technology for the design of these components is inadequate.

*Design Study of Large Unconventional Liquid Propellant Rocket Engines and Vehicles, Aerojet-General Report R-LRP-257, 18 April 1962 (c).
Study of Large Unconventional Liquid Propellant Rocket Engines and Vehicles, North-American Aviation Report R-3517D, 30 April 1962 (c).

V, Introduction (cont.)

B. PROGRAM SCOPE

1. Engine-Systems Considered

To cover the family of future launch-vehicle engine systems, the following engine-design parameters were considered:

a. Thrust, 1.5, 6, and 24 x 10^6 lb

b. Pump discharge pressure, 2000, 4000, and 6000 psia (nominal)

c. Propellants, LO_2/LH_2 and LO_2/RP-1

d. Engine cycle, gas generator and staged combustion

e. Engine configuration, single engine, multiple chamber — single pump, and modular engine

2. Auxiliary Components Studied

A pump-fed liquid rocket engine may be considered as an assembly of several primary components, or subassemblies, and numerous auxiliary components. The primary components are the turbopump assembly (TPA); the pump-drive assembly (PDA); the gas generator(s), or primary combustor(s); the thrust chamber assembly (TCA), or secondary combustor; and the expansion section. The remainder of the engine consists of auxiliary systems and components.

The auxiliary systems include the thrust vector control (TVC), propellant-utilization (PU), and tank-pressurization systems. The auxiliary components are all of those items that interconnect the primary components and comprise the auxiliary systems. In some instances, an auxiliary component may be an integral part of a primary component, e.g., a ring-gate pump-discharge valve located in the pump housing between the impeller and the diffuser.

For the purpose of this investigation, only those auxiliary component designs directly influenced by the engine thrust level and system operating

V, B, Program Scope (cont.)

pressures were considered. The components evaluation included the flow-control valves, lines and ducts with required flexible joints, and heat exchangers. Instrumentation, small lines and tubes, solenoid valves, pilot valves, check valves, and pressure switches (although pressure-affected) were not included because their design is generally not a function of engine thrust level and may be considered to be within the capability of present technology.

Most of the auxiliary components studied were those subject to high pressure during engine operation. The exceptions were certain suction (feed) ducts and valves that were evaluated to establish the relative severity of large-size requirements.

3. Basic Assumptions

The following assumptions were made early in this program to provide uniform ground rules for the entire investigation.

a. All engine systems considered are designed for application to a launch vehicle of 24×10^6-lb of thrust at sea level. Three examples of this vehicle are shown in Figure V-B-1.

b. Nominal and maximum peak accelerations of 6 and 10 g, respectively, were selected as being compatible with the potential missions of such a vehicle.

c. Connectors and seals (static, sliding, or rotating) are the primary subject of current efforts of other agencies (Appendix A); detailed investigation of these problem areas is not within the scope of this study. However, the establishment of seal and connector design requirements, such as size and pressure, is a definite objective of this program.

V, Introduction (cont.)

C. APPLICATION FACTORS

In order that the program effort would be directed along realistic guidelines and that the results would be significant, the probable influence of auxiliary components on the overall launch-vehicle-system performance was studied. To provide the required information, several hypothetical launch-vehicle missions were analyzed.

While the engines studies conducted under this program are applicable to a wide variety of booster missions, the single-stage-to-300-NM-orbit mission was considered for the auxiliary-component evaluation because it imposes severe design requirements on the propulsion system.

A vehicle with a total lift-off weight of 19.2×10^6 lb and a sea-level thrust of 24×10^6 lb will insert a payload of approximately 960,000 lb into such an orbit if the engine employs high chamber pressure. A similar 24×10^6-lb-thrust system using less-efficient engines of current technology (low chamber pressure) would deliver a payload of approximately 670,000 lb into a 300-NM orbit.

An analysis of existing detailed designs of high- and low-pressure LO_2/LH_2 engines revealed that 14.0% of the total high-pressure-engine weight is in the auxiliary components, as compared with 17.6% for the low-pressure engine. If the weight of the auxiliary components were reduced to zero for both engine systems, the payload of the low-pressure system could be increased by 9.94% as compared with 5.25% for the high-pressure system. Therefore, for launch-vehicle engines, a 10% reduction in auxiliary component weight would only increase the payload of high-pressure systems by approximately 1/2% and low-pressure systems by approximately 1%.

It was concluded that weight-saving, while desirable and to be considered in all component studies, may not offer sufficient gain to warrant advanced-technology work for this purpose alone. Therefore, emphasis in this program has been on those problem areas where severity threatens to preclude the design of functionally-adequate auxiliary components.

V, Introduction (cont.)

D. APPROACH

The initial effort was the selection of a representative group of engines with variations in thrust level, cycle, propellant, and configuration, as previously described. Preliminary designs were prepared for each of the selected engine systems; steady-state and transient analyses were performed to establish component functional environmental requirements.

Representative auxiliary components providing the most critical design problems were selected from each of these engines. Because general design concepts rather than detailed component designs were required, a preliminary design analysis was performed for each of these representative components. Wherever possible, thrust level and pressure were treated parametrically in the analysis.

An investigation of materials for component application provided verification of the suitability of the structural metals under consideration, and created interest in the application of filament-wound reinforced-plastics technology to high-pressure lines. A study of seal materials and designs, initiated in the course of this investigation, was cancelled when it became known that more comprehensive efforts were being conducted under the auspices of other contracts.

During the course of this program, a literature survey was conducted to ensure that all available, pertinent information was considered. The reviewed literature is listed in the bibliography, Appendix I.

On completion of the engine system and component studies, a critical review was made of the various component-design requirements and the difficulties encountered while preparing preliminary designs to meet these requirements. This review has served to identify areas of required advanced-technology effort.

V, D, Approach (cont.)

Finally, programs have been outlined describing the necessary advanced-technology programs that must be performed to enable actual flight hardware to be designed on the basis of sound, experimentally-verified data rather than by extrapolating existing empirical data.

E. RESULTS

The results of this program are presented in detail in the sections pertaining to engine-systems analysis, component-design criteria, required technology, and recommended programs. A brief summary of the more significant results includes the following:

1. Engine Systems

There appears to be no reason why an engine in the later stages of development should induce peak pressures in the high-pressure components that exceed nominal operating pressures by more than 10%. For this reason, the maximum operating pressure considered in this program for component-design purposes was 6600 psia. In practice, most engines will never reach 10% overpressure; but a $\pm 5\%$ tolerance on pump discharge pressure (Section VI,B,2,a) combined with the effects of vehicle acceleration (Section VI,B,2,b) and water-hammer (Section VI,C,3), or shutdown-transient effects (Section VI,C,2), might result in a few engines reaching such pressure levels on occasion.

Propellant valves downstream of the pump normally must be capable of sealing tightly against the static head of propellant in the engine, suction lines, and tank, but are at least partially open during the periods of high-pressure pump operation. These propellant valves, therefore, need to be designed for "zero" or minimal leakage only at relatively low pressures. Internal seals will experience the high pressures, but some leakage there can usually be tolerated. High-pressure valves (e.g., TVC valves that must seal against high pressure) present the most severe problems.

V, E, Results (cont.)

Valve opening and closing elapsed time for start and shutdown are much the same (in the order of 1/2 to 2 sec) for most engines regardless of thrust level, cycle, propellant, configuration, or pressure. At the velocities required for the moving elements of the valve, inertial forces are still outweighed by friction forces by approximately 100:1.

Start-transient studies have indicated that practically any engine, regardless of cycle, propellant, thrust level, or pressure, may be started satisfactorily with linear actuation of the pump discharge or primary and secondary combustor valves, by adjustment made only to the total time for each valve to open. No variable actuation rate or special valve resistance versus stroke characteristic is required for simple startup or shutdown.

Steady-state analysis of engines of various system pressures revealed that the higher pressure engines are less sensitive to pressure drop through the auxiliary components and, therefore, can tolerate somewhat smaller line and valve diameters with their correspondingly higher resistances.

The only application where a hot-gas valve (or any large valve) may be required to seal against high pressure is in a hot-gas secondary injection thrust-vector-control system. The nature of such a system may well permit a slight leakage in the closed position, but the application is considerably more severe than other hot-gas valve applications (such as a turbine-bypass propellant-utilization valve). On engine systems with more than eight discrete combustion chambers, the thrust-vector-control valve is the largest high-pressure valve on the engine, if fluid injection is used.

Large launch vehicles with the LO_2 tank on top have a severe increase in oxidizer-pump net positive suction head (NPSH) as the vehicle accelerates. This condition necessitates either throttling with the oxidizer suction valve, throttling the oxidizer-pump inducer (by throttling the high-pressure LO_2 recirculated to the

V, E, Results (cont.)

inducer turbine), controlling the oxidizer-pump head-rise within the pump, or providing the pump discharge valve (or thrust chamber and gas generator valves) with modulating capability. Such valve-modulating capability may also be required if the engine system is to be throttled to a reduced thrust level.

2. Auxiliary Components

The largest high-pressure valves, lines, and flexible joints on engines of 1.5 to 24×10^6-lb of thrust may be expected to range in size from 7 to 20 in. in diameter.

There are no problems associated with the design of high-pressure auxiliary components that cannot be solved or circumvented by a "brute force" approach using present technology. The design of efficient auxiliary component hardware would be made possible by the results of the recommended advanced-technology programs.

The design of high-pressure valves for specific application requires that not just one, but a variety of valve concepts be available. The variety of engine requirements may suggest the use of sleeve, poppet, venturi, or ball valves in either inline or angle configurations to satisfy particular functional requirements and envelope constraints.

Sealing remains the primary problem in the design and development of high-pressure valves. The present limitations on seal technology may dictate the selection of the type of valve for a particular application. The present lack of adequate sliding seal technology makes the poppet valve (possibly with a welded-bellows interior seal) the most promising for development in the immediate future.

The staged-combustion cycle may provide requirements for modulating capability in the high-pressure discharge valves to perform thrust control, thrust modulation, or propellant-utilization functions. This approach, while minimizing

V, E, Results (cont.)

the number of valves on the engine, places the more stringent demand of modulating capability on the large valves used simply for on-off service on current gas-generator cycle engines.

The design of propellant and hot-gas lines and ducts with diameters up to 20 in. and internal pressures up to 6600 psi requires more comprehensive structural analysis than has been required for smaller, low-pressure engines. This task is simplified by the use of currently available computer programs for piping-system analysis.

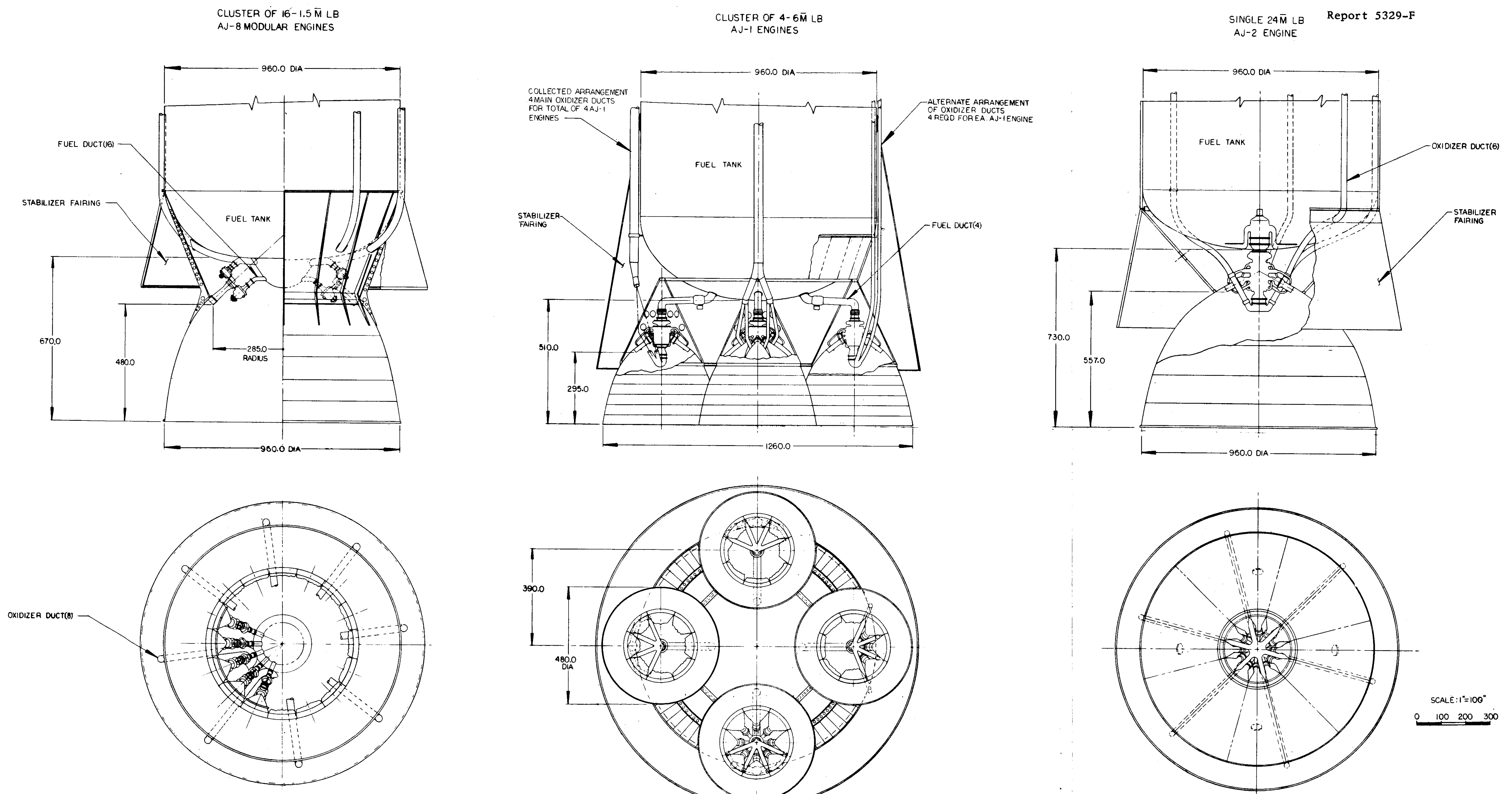
The flexibility requirements for large-diameter, high-pressure lines demands flexible joints using formed bellows up to 20 in. in diameter for proof pressures from 7900 to 13,400 psi and capable of gimbaling or bending from ± 7 to $\pm 12°$. By comparison, this investigation has shown that bellows currently manufactured for this size range are for proof pressures of less than 1000 psi.

Some bellows manufacturers contend that the current technology can be extrapolated for the design of bellows joints for these large-diameter, high-pressure applications, but that fabrication development would be required. A demonstration program to design, fabricate, and test several such bellows joints is needed to verify the feasibility of developing hardware for specific applications and to provide criteria for establishing limits of the various bellows parameters such as number of plies, ply thickness, spring rate, and convolution height, pitch, number, and configuration.

A significant potential exists in the use of filament-wound reinforced-plastic lines to provide up to one-hundred times the flexibility of a metal line without flexible joints. Such a line would have a weight approximately one-half that of the continuous metal line. A feasibility program is recommended to verify the potential of filament-wound lines and to investigate solutions to liner, connector, flame-protection, and fabrication problems.

V, E, Results (cont.)

The most probable application of heat exchangers to these rocket engines is for tank pressurization. No problems are visualized that would require advanced-technology work to provide efficient heat exchangers for these engines. The high-pressure fluid media require that careful consideration be given to location of the heat exchanger in the engine system in order that minimum tube wall thickness may be utilized.



24M-1b Vehicles

Figure V-B-1

VI. ENGINE-SYSTEMS STUDIES

A. PRELIMINARY ENGINE EFFORT

1. Selection of Engine Systems

To carry out an effective and knowledgeable study of auxiliary components and their relationship to effects of size and pressure in liquid rocket engines, it is necessary to select a realistic group of engines from which to derive the components and their requirements. Consideration of a proper collection of engines simplifies the selection of components to be analyzed, and at the same time, provides the most useful and applicable set of criteria for the auxiliary components.

There are many conditions critical to the design and specification of the auxiliary components, and each had to be considered at some point in the program. These conditions include size (based on flow considerations and envelope limitations), system pressure and temperature, flexure and/or reaction load requirements, response times for controls, acceleration and shock, and vibration. The basic engine parameters having the greatest influence on these conditions are:

a. Thrust level

b. Engine cycle

c. Propellants

d. Engine configuration: number of thrust chambers and pumps

e. Thrust-chamber and pump discharge pressure

Three thrust levels are considered: 1.5, 6, and 24×10^6-lb. This adequately covers the thrust range of engines anticipated for future space missions. In addition to thrust, the following variations are considered: gas-generator and

VI, A, Preliminary Engine Effort (cont.)

staged-combustion engine cycles; LO_2/LH_2 and LO_2/RP-1 propellant combinations; and multiple thrust chamber with a single pump and single thrust chamber with a single pump engine configurations. The latter configuration at the 1.5 x 10^6-lb thrust level constitutes a multiple thrust chamber, multiple pump configuration when several modules are clustered together in a common nozzle. Pressure variations are considered at the component level; therefore, nominal chamber pressures of 2500 psia for the staged-combustion-cycle engines and 3370 psia for the gas-generator-cycle engines were used for design and layout work (the 3370 psia requires approximately the same pump-discharge pressure as the staged-combustion engines). This approach permitted maximum utilization of existing design experience when this program began.

The following matrix illustrates the 24 possible combinations of the above engine parameters at equivalent pressure levels. The engine designations (AJ-1, AJ-2, etc.) were assigned simply to facilitate identification of the various conceptual designs studied and have no connection with similar designations of actual Aerojet engines, past or future. These engines are described later in this section.

1.5 x 10^6 lb		6 x 10^6 lb		24 x 10^6 lb		Thrust Level	
S.C.*	M.C.	S.C.	M.C.	S.C.	M.C.	Engine Configuration / Propellants	Cycle
		AJ-5				LO_2/LH_2	Gas Generator
						LO_2/RP-1	
AJ-8***		AJ-6	AJ-1		AJ-2	LO_2/LH_2	Staged Combustion**
			AJ-3			LO_2/RP-1	

* S.C., Single Chamber; M.C., Multiple Chamber

** Staged combustion refers to that cycle wherein all or a major portion of one propellant is partially burned with a small portion of the other propellant in a primary combustor to provide working fluid to drive the gas turbine(s). This exhaust is then burned with the remainder of the second propellant in the combustion chamber, or secondary combustor.

*** The AJ-8 cycle differs in that it is a gas/gas staged-combustion cycle (a fuel-rich primary combustor providing gas for the fuel-pump turbine and an oxidizer-rich primary combustor providing gas for the oxidizer-pump turbine), as compared with the gas/liquid staged-combustion cycle described above.

VI, A, Preliminary Engine Effort (cont.)

To satisfy the program objectives within the scope of the contract, six basic engines were selected for design and analysis. These six engines were chosen such that all major parameter variations could be investigated.

By selecting a baseline engine and then evaluating parameter variations, all 24 combinations may be indirectly considered. The shaded area in the matrix illustrates this philosophy (with AJ-1 being the baseline engine). Engine AJ-1 is a 6 x 10^6-lb-thrust, staged-combustion cycle, LO_2/LH_2, single-pump, multiple thrust-chamber engine. Engine AJ-3 was selected to show propellant variation (LO_2/RP-1); Engine AJ-2, change in thrust level (24 x 10^6 lb); Engine AJ-6, change in configuration by using a single combustion chamber; Engine AJ-8, change in configuration when a single pump is used with a single chamber in a staged-combustion engine; and Engine AJ-5 to investigate the problems of large valves and lines that would be associated with a gas-generator-cycle engine in contrast to AJ-6.

In making these engine selections, it was considered that (1) a 24 x 10^6-lb-thrust single chamber is unlikely for a next-generation engine, (2) single engines were not selected for design and analysis where a multiple combustor engine of similar thrust-chamber size had been selected, and (3) variations in engine cycle for similar engines were not considered for both propellant systems because no unique auxiliary component problems are solely dependent on the propellant.

2. Specification of Engine-Operation Conditions

The specifications for the basic engines selected in the previous section were established in the following manner. Thrust-chamber pressure, thrust-chamber mixture ratio, and nozzle-area ratio were selected on the basis of optimum overall vehicle performance. These values differ with propellant, cycle, and nozzle and are shown in the tables of engine specifications in Section VI,A,3. Specific impulse is determined from thermochemical data based on the above factors and the

VI, A, Preliminary Engine Effort (cont.)

combustion and nozzle efficiencies (98 and 94.8%, respectively) established by the hot-flow and cold-flow tests of thrust chambers and nozzles. The primary combustor mixture ratio was determined by the limiting turbine-inlet temperature (1900°R with LO_2/LH_2 and 1773°R with LO_2/RP-1*). Flow rates throughout the engine system were then calculated.

The vehicle tankage was sized on the basis of a single-stage-to-300-NM-orbit mission; this presents the most severe pump-inlet conditions (see Section VI,B,2,b). Pump-inlet conditions were determined by measuring the static head due to the height of the tank, the vapor pressure of the propellant, and the tank pressurization necessary to meet the minimum suction-pressure requirements for sound pump design. Based on previous experience and detailed calculations, pump-discharge-pressure requirements were established from a pump-turbine power balance (using the established efficiencies) and a schedule of valve, line, and injector pressure drops. These pressure drops vary from engine to engine and are summarized in the tables of Section VI,A,3. The pump efficiencies were established by specifying a pump specific speed corresponding to the maximum efficiency for each pump type (1500 to 3000 for LO_2 and 800 to 1800/stage for LH_2). The various other pump-operating parameters were calculated by the following expressions:

$$N_s = \frac{N\,Q^{1/2}}{H^{3/4}} \quad \text{and} \quad S = \frac{N\,Q^{1/2}}{(NPSH)^{3/4}}$$

where:

N_s-Pump specific speed, $\frac{rpm\ (gpm)^{1/2}}{(ft)^{3/4}}$

N-Pump speed, rpm

Q-Volumetric flow rate, gpm

H-Pump head, ft

S-Suction specific speed, $\frac{rpm\ (gpm)^{1/2}}{(ft)^{3/4}}$

NPSH-Net positive suction head, ft

*Higher temperatures could be achieved with cooled turbine blades but it was not believed necessary to assume this technology for this study. The particular values used in this study do not necessarily reflect true limits; rather, they are values selected to permit maximum use of existing data.

VI, A, Preliminary Engine Effort (cont.)

The pump rotating speeds were selected on the basis of previous rotating machinery design experience. The turbine efficiencies were established on the basis of the degree of reaction suited to the approximate pressure ratio desired and the U/C_o ratio, where U = blade-tip speed in ft/sec and C_o is the isentropic spouting velocity in ft/sec. The efficiencies range from 0.66 to 0.80. These specifications sufficiently defined the engine operating characteristics for the purpose of further analysis; therefore, no impeller or blade design was necessary.

3. Description of Engine Systems

Selection of the specific components to be investigated in the component-design portion of this contract was made from the six basic engines. The engine layouts show the concepts of the engines in detail and establish such specifications as length of lines and available locations for valves and flexible joints. They do not, however, show line diameters to scale; line diameters were determined later in the program.

a. Engine AJ-1 (Baseline)

This baseline engine has a 6×10^6-lb thrust level, uses LO_2/LH_2 propellants, multiple thrust-chambers, a single-shaft pumping system, and the liquid/gas staged-combustion cycle in a forced-deflection nozzle. The engine operates at a thrust-chamber pressure of 2500 psia. The pump discharge pressures are 4100 psia for LO_2 and 4550 psia for LH_2. Much of the difference in pump discharge pressure between the two pumps is caused by the coolant-jacket pressure drop in the LH_2 circuit that is in series with the primary combustor — turbine circuit. An engine flow schematic is shown in Figures VI-A-1. Engine layouts for the AJ-1 engine are shown in Figures VI-A-2 through VI-A-5. Specifications and pressure schedules are shown in Tables VI-A-1 and -2.

VI, A, Preliminary Engine Effort (cont.)

b. Engine AJ-2

Two versions (AJ-2A and AJ-2B) of this engine have been designed. Both are 24 x 10^6 -lb-thrust engines but differ slightly in their application of the staged-combustion cycle. AJ-2A is identical to AJ-1 in basic configuration and engine parameters, but differs in thrust level, number of thrust chambers, and pressure schedule. AJ-2B has significantly different auxiliary component requirements and engine specifications. It utilizes only 30% of the liquid hydrogen flow to cool the thrust chamber as compared with 100% in AJ-2A. Engine flow schematics and layouts for AJ-2A and AJ-2B are shown in Figures VI-A-6 through VI-A-11. Specifications and pressure schedules are shown in Tables VI-A-1, -3 and -4.

c. Engine AJ-3

This engine is a 6 x 10^6 -lb-thrust, LO_2/RP-1, multiple thrust-chamber, single pump system, staged-combustion cycle engine that operates at a thrust-chamber pressure of 2500 psia. The pump discharge pressures for this engine are 4000 psia for both LO^2 and RP-1 pumps. The engine flow schematic and the basic layout are shown in Figures VI-A-12 and -13. The pressure schedule and specifications are shown in Tables VI-A-1 and -5.

d. Engine AJ-5

AJ-5 is a 6 x 10^6 -lb-thrust, LO_2/LH_2, single thrust-chamber, single pumping-system (however, separated pumps in this case), gas-generator cycle engine that operates at a thrust-chamber pressure of 3370 psia. This pressure is achieved with the same nominal pumps utilized in the other engines, but with the nonseries combustion of the gas-generator cycle a higher thrust-chamber pressure is achieved. The pump discharge pressures are 4100 psia for both pumps. The engine flow schematic and basic layout are shown in Figures VI-A-14 and -15. The pressure schedule and specifications are shown in Tables VI-A-1 and -6.

VI, A, Preliminary Engine Effort (cont.)

e. Engine AJ-6

This engine is identical to AJ-1 except that the TPA is mounted on a single thrust chamber with a deLaval nozzle. The nozzle-area ratio was adjusted so that sea-level specific impulse is the same as that of the forced-deflection nozzle AJ-1. Consequently, the flow schematic, specifications, and pressure schedule are identical to those of AJ-1. A layout of AJ-6 is shown in Figure VI-A-16.

f. Engine AJ-8

Engine AJ-8 is a 1.5×10^6 -lb-thrust, LO_2/LH_2, single thrust-chamber, single pumping-system (again separated) gas/gas staged-combustion cycle engine. The cycle is gas/gas in the sense that all of the propellant except the coolant is combusted in the primary combustors. The engine operates with a fuel-rich primary combustor driving the fuel turbine and an oxidizer-rich primary combustor driving the oxidizer turbine. The engine operates at a thrust-chamber pressure of 2500 psia with pump discharge pressures of 4930 psia for both pumps. The engine flow schematic and basic layouts are shown in Figures VI-A-17 to -19. The two configurations of the engine shown differ in the orientation of the pumps and the resultant auxiliary component requirements. The pressure schedule and specifications are shown in Tables VI-A-1 and -7.

g. AJ-2000 and AJ-6000

To establish reasonable pressure schedules covering the whole range of pump discharge pressures specified for the program, high- and low-pressure versions of AJ-1 were analyzed. The resultant pressure schedules are shown in Tables VI-A-1, -8 and -9.

TABLE VI-A-1

ENGINE SPECIFICATIONS

		AJ-1	AJ-2A	AJ-2B	AJ-3	AJ-5	AJ-8
ENGINE							
THRUST (F)	LB	6×10^6	24×10^6	24×10^6	6.0×10^6	6×10^6	1.5×10^6
SPECIFIC IMPULSE (I_{SP})	SEC	383	383	383	303	383.6	371
MIXTURE RATIO (MR)	- -	6.0	6.0	6.0	2.6	5.45	6.0
THRUST CHAMBER							
THRUST (F)	LB	6.0×10^6	24×10^6	24×10^6	6×10^6	5.87×10^6	1.5×10^6
CHAMBER PRESSURE, PLENUM (P_C)	PSIA	2500	2500	2500	2500	3372	2500
CHAMBER PRESSURE INJ. FACE (P_J)	PSIA	2580	2580	2580	2580	3480	2580
SPECIFIC IMPULSE (I_{SP}) TC	SEC	383	383	383	303	389	371
THRUST COEFFICIENT (C_F)	- -	1.615	1.615	1.615	1.553	1.606	1.593
CHARACTERISTIC VELOCITY (C^*)	FT/SEC	7630	7630	7630	6278	7792	7494
EFFECTIVE EXHAUST VELOCITY (U_E)	FT/SEC	12322	12323	12323	10310	12517	11936
EXIT PRESSURE (P_C)	PSIA	14.7	14.7	14.7	14.7	1.187	14.7
PRESSURE RATIO (P_C/P_e)	- -	170	170	170	170	2845	170
TOTAL FLOW RATE ($\dot{W}_T$)	LB/SEC	15666	62663	62663	19799	15088.0	4044
MIXTURE RATIO (MR_{TC})	- -	6.0	6.0	6.0	2.6	6.0	6.0
FUEL FLOW RATE ($\dot{W}_F$)	LB/SEC	2238	8952	8952	5500	2155.4	578
OXIDIZER FLOWRATE ($\dot{W}_O$)	LB/SEC	13428	53711	53711	14299	12932.6	3466
THROAT AREA (A_T)	IN^2	100/CH	488/CH	488/CH	179/CH	1050	365
AREA RATIO (ϵ)	- -	120	120	120	100	40	120
NOZZLE EFFICIENCY (λ)	- -	.948	.948	.948	.965	.948	.925
COMBUSTION EFFICIENCY (η_C)	- -	.98	.98	.98	.98	.98	.98

TABLE VI-A-1 (cont.)

		AJ-1		AJ-2A		AJ-2B		AJ-3		AJ-5		AJ-8	
		FUEL	OXIDIZER	FUEL	OXIDIZER	FUEL	OXIDIZER	FUEL	OXIDIZER	FUEL	OXIDIZER	FUEL	OXIDIZER
MAIN STAGE PUMP													
Flow Rate ($\dot{W}$)	LB/SEC	2238	15578	8952	60651	8951	59552	5500	16389	2421	14796	578	3854
Discharge Pressure (P_D)	PSIA	4550	4100	4679	4001	4900	4800	4000	4000	4100	4100	4931	4931
Specific Speed (N_s)	RPM (GPM)½ / (FT)3/4	429	2554	372	2214	360	1944	1424	2759	489	2557	407	1460
Volumetric Flow (Q)	GPM	228,276	99,169	913,092	386,103	913,095	379,106	49,315	104,332	246,984	94,189	58,914	24,532
Total Head Rise (ΔH)	FT	147,853	7869	152046	7948	159263	9344	11234	7668	133032	7925	160,509	9627
Shaft Speed (N)	RPM	6776	6776	3000	3000	3000	3000	7000	7000	6861	6999	13,464	9072
Suction Specific Speed (S)	RPM (GPM)½ / (FT)3/4	30,000	21,228	25148	21206	25148	19804	23258	22630	27287	23495	42509	15719
Net Positive Suction Head (NPSH)	FT	513	467	553	391	553	423	271	646	624.7	411.9	326	405
Propellant Density (ρ)	LB/FT^3	4.4	70.5	4.4	70.5	4.4	70.5	50.05	70.5	4.4	70.5	4.4	70.5
Propellant Temp. (T)	^oF	-423	-293	-423	-293	-423	-293	80	-293	-422.5	-293.4	-422.5	-293.4
Efficiency (η_p)	-	0.72	0.72	.698	.725	.718	.728	.703	.673	.729	.701	.806	.662
HYDRAULIC INDUCER													
Flowrate ($\dot{W}_I$)	LB/SEC		13428		.53711		,53711		14299		13212		3465
Total Head Rise (ΔH_I)	FT		422		361		393		364		381.7		375
Specific Speed (N_s)	RPM (GPM)½ / (FT)3/4		5170		4922		4502		7323		5165		5137
Shaft Speed (N)	RPM		1649		703		683		2021		1590		3010
Net Positive Suction Head (NPSH)	FT		45		30		30		100		30.1		30.7
Suction Specific Speed (S)	RPM (GPM)½ / (FT)3/4		27,645		31981		31084		19289		35871		34258
Efficiency (η_I)			.588		.538		.651		.597		.577		.572

TABLE VI-A-1 (cont.)

		AJ-1	AJ-2A	AJ-2B	AJ-3	AJ-5		AJ-8	
						OXIDIZER	FUEL	OXIDIZER	FUEL
INDUCER TURBINE									
FLOWRATE ($\dot{W}_{IT}$)	LB/SEC	2150	6940	5841	2090	1584		388	
PERCENT OF PROP. FLOW	- -	16.0	12.9	10.9	14.6	12.0		11.2	
EFFICIENCY (η_{IT})	- -	.570	.653	.594	.543	.696		.608	
SHAFT HORSEPOWER (SHP_{IT})	HP	17,544	65480	58904	15835	15889		4129	
GAS TURBINE									
MIXTURE RATIO (MR_{PC})	- -	.915	.915	1.05	38.0	1.05	1.05	110	.90
PROPELLANT FLOW ($\dot{W}_T$)	LB/SEC	4285	17142	12829	14675	545	545	3134	760
RATIO OF SPECIFIC HEATS (k)	- -	1.35	1.35	1.35	1.298	1.35	1.35	1.293	1.363
GAS CONSTANT (R)	$\frac{FT\text{-}LB}{LB°R}$	396	396	396	48.5	395.7	395.7	51.5	399
INLET TEMPERATURE (T_{TI})	°F	1440	1440	1440	1313	909	1440	1141	1160
CHARACTERISTIC VELOCITY (C^*)	FT/SEC	7273	7273	7274	2493	7274	7274	2445	6720
SPECIFIC HEAT (C_P)	$\frac{FT\text{-}LB}{LB°R}$	1526	1526	1526	211	1526.3	1526.3	227	1497
EFFICIENCY (η_T)	- -	.822	.746	.691	.789	.707	.660	.673	.731
INLET PRESSURE (P_{TI})	PSIA	3496	3620	4270	3590	427	3570	4192	4192
PRESSURE RATIO	- -	1.279	1.326	1.547	1.329	2.455	8.357	1.397	1.397
EXIT PRESSURE (P_{TC})	PSIA	2734	2730	2760	2700	174	427	3000	3000
SHAFT HORSEPOWER (SHP)	HP	1,146,642	4,753,222	5,001,420	499,134	304309	803201	101,825	209,004

TABLE VI-A-2

AJ-1 ENGINE PRESSURE SCHEDULE

	Circuit*		
	Oxidizer Primary Combustor	Fuel	Oxidizer Secondary Combustor
Pump Discharge	4100	4550	4100
Δ P, Line	50	50	50
Δ P, Pump Discharge Valve	50	50	50
Coolant Jacket Inlet	----	4450	----
Δ P, Coolant Jacket	----	350	----
Orifice Inlet	----	----	4000
Δ P, Orifice and Valve	----	----	1120
Primary Combustor, Injector Inlet	4000	4100	----
Δ P, Primary Combustor Injector	250	350	----
Primary Combustor Injector Face	3750	3750	----
Primary Combustor Plenum	3621	3621	----
Pressure Ratio, Primary Combustor to Turbine Nozzle	----	----	----
Δ P, Primary Combustor to Turbine Nozzle	125	125	----
Turbine Nozzle Inlet	3496	3496	----
Turbine Pressure Ratio	1.279	1.279	----
Δ P, Turbine	762	762	----
Turbine Exit	2734	2734	----
Secondary Combustor Injector Inlet	2734	2734	2880
Δ P, Secondary Combustor Injector	154	154	300
Secondary Combustor Injector Face	2580	2580	2580
Secondary Combustor Plenum	2500	2500	2500

* Pressures are in psia; pressure drops are in psi.

TABLE VI-A-3

AJ-2A ENGINE PRESSURE SCHEDULE

	Circuit*		
	Oxidizer Primary Combustor	Fuel	Oxidizer Secondary Combustor
Pump Discharge	4101	4679	4101
ΔP, Line	46	50	50
ΔP, Pump Discharge Valve	51	49	51
Coolant Jacket Inlet	----	4580	----
ΔP, Coolant Jacket	----	492	----
Orifice Inlet	----	----	4000
ΔP, Orifice and Valve	----	----	1126
Primary Combustor Injector Inlet	4004	4088	----
ΔP, Primary Combustor Injector	254	338	----
Primary Combustor Injector Face	3750	3750	----
Primary Combustor Plenum	3650	3650	----
ΔP, Primary Combustor to Turbine Nozzle	30	30	----
Turbine Nozzle Inlet	3620	3620	----
Turbine Pressure Ratio	1.326	1.326	----
ΔP, Turbine	890	890	----
Turbine Exit	2730	2730	----
Secondary Combustor Injector Inlet	2730	2730	2874
ΔP, Secondary Combustor Injector	150	150	294
Secondary Combustor Injector Face	2580	2580	2580
Secondary Combustor Plenum	2500	2500	2500

* Pressures are in psia; pressure drops are in psi.

TABLE VI-A-4

AJ-2B ENGINE PRESSURE SCHEDULE

	Circuit*			
	Fuel P.C.**	Fuel S.C.***	Oxid P.C.	Oxid S.C.
Pump Discharge	4900	4900	4800	4800
ΔP, Line	50	50	50	50
ΔP, Pump Discharge Valve	50	50	50	50
Coolant Jacket Inlet	----	4800	----	----
ΔP, Coolant Jacket	----	500	----	----
Orifice Inlet	----	4300	----	----
ΔP, Orifice	----	1470	----	----
Primary Combustor Injector Inlet	4800	----	4700	----
ΔP, Primary Combustor Injector	350	----	250	----
Primary Combustor Injector Face	4450	----	4450	----
Primary Combustor Plenum	4325	----	4325	----
ΔP, Primary Combustor to Turbine Nozzle	55	----	55	----
Turbine Nozzle Inlet	4270	----	4270	----
Turbine Pressure Ratio	1.547	----	1.547	----
ΔP, Turbine	1510	----	1510	----
Turbine Exit	2760	----	2760	----
Orifice Inlet	----	----	----	4706
ΔP, Orifice and Valve	----	----	----	1795
Secondary Combustor Injector Inlet	2760	2830	2760	2905
ΔP, Secondary Combustor Injector	180	250	180	325
Secondary Combustor Injector Face	2580	2580	2580	2580
Secondary Combustor Plenum	2500	2500	2500	2500

*Pressures are in psia; pressure drops are in psi.

**Primary Combustor

***Secondary Combustor

TABLE VI-A-5

AJ-3 ENGINE PRESSURE SCHEDULE

	Circuit*			
	Oxid S.C.**	Oxid P.C.***	Fuel P.C.	Fuel S.C.
Pump Discharge	4000	4000	4000	4000
ΔP, Line	50	50	50	50
ΔP, Pump Discharge Valve	50	50	50	50
Coolant Jacket Inlet	----	----	----	3900
ΔP, Coolant Jacket	----	----	----	500
Orifice Inlet	----	----	----	3400
Primary Combustor Injector Inlet	----	3900	3900	----
ΔP, Primary Combustor Injector	----	150	150	----
Primary Combustor Injector Face	----	3750	3750	----
Primary Combustor Plenum	----	3625	3625	----
ΔP, Primary Combustor to Turbine Nozzle	----	35	35	----
Turbine Nozzle Inlet	----	3590	3590	----
Turbine Pressure Ratio	----	1.330	1.330	----
ΔP, Turbine	----	890	890	----
Turbine Exit	----	2700	2700	----
Orifice Inlet	3900	----	----	3400
ΔP, Orifice	1070	----	----	570
Secondary Combustor Injector Inlet	2830	2700	2700	2830
ΔP, Secondary Combustor Injector	250	120	120	250
Secondary Combustor Injector Face	2580	2580	2580	2580
Secondary Combustor Plenum	2500	2500	2500	2500

*Pressures are in psia; pressure drops are in psi.

**Secondary Combustor

***Primary Combustor

TABLE VI-A-6

AJ-5 ENGINE PRESSURE SCHEDULE

	Circuit*			
	Fuel G.G.**	Fuel T.C.***	Oxid G.G.	Oxid T.C.
Pump Discharge	4100	4100	4100	4100
Δ P, Line	50	50	50	50
Δ P, Valve	50	50	50	50
Coolant Jacket Inlet	----	4000	----	----
Δ P, Coolant Jacket	----	370	----	----
Gas Generator Injector Inlet	4000	----	4000	----
Δ P, Gas Generator Injector	250	----	250	----
Gas Generator Plenum	3750	----	3750	----
First Turbine Inlet	3570	----	3570	----
Δ P, First Turbine	3143	----	3143	----
First Turbine Pressure Ratio	8.357	----	8.357	----
Second Turbine Inlet	427	----	427	----
Δ P, Second Turbine	253	----	253	----
Second Turbine Pressure Ratio	2.455	----	2.455	----
Second Turbine Exhaust	174	----	174	----
Orifice Inlet	----	----	----	4000
Δ P, Orifice	----	----	----	320
Thrust Chamber Injector Inlet	----	3630	----	3680
Δ P, Injector	----	150	----	200
Injector Face	----	3480	----	3480
Thrust Chamber Plenum	----	3372	----	3372

*Pressures are in psia; pressure drops are in psi.

**Gas Generator

***Thrust Chamber

TABLE VI-A-7

AJ-8 ENGINE PRESSURE SCHEDULE

	Circuit*		
	Fuel Secondary Combustor	Fuel Primary	Oxidizer
Pump Discharge	4930	4930	4930
ΔP, Line	30	30	65
ΔP, Secondary Combustor Valve (SCV)	30	----	----
Coolant Jacket Inlet	4870	----	----
ΔP, Coolant Jacket	1500	----	----
Orifice Inlet	----	----	4865
ΔP, Orifice	----	----	15
Primary Combustor Valve (PCV) Inlet	----	4900	4850
ΔP, Primary Combustor Valve (PCV)	----	120	70
Primary Combustor Injector Inlet	----	4780	4780
ΔP, Primary Combustor Injector	----	180	180
Primary Combustor Injector Face	----	4600	4600
Primary Combustor Plenum	----	4490	4490
Pressure Ratio, Primary Combustor to Turbine Nozzle	----	1.07	1.07
ΔP, Primary Combustor to Turbine Nozzle	----	298	298
Turbine Nozzle Inlet	----	4192	4192
Turbine Pressure Ratio	----	1.397	1.397
ΔP, Turbine	----	1192	1192
Turbine Exit	----	3000	3000
Orifice Inlet	3370	----	----
ΔP, Orifice	390	----	----
Pressure Ratio, Turbine Exit to Injector	----	1.057	1.057
ΔP, Turbine Exit to Injector	----	160	160
Secondary Combustor Injector Inlet	2980	2840	2840
ΔP, Secondary Combustor Injector	400	260	260
Secondary Combustor Injector Face	2580	2580	2580
Secondary Combustor Plenum	2500	2500	2500

*Pressures are in psia; pressure drops are in psi.

TABLE VI-A-8

AJ-2000 ENGINE PRESSURE SCHEDULE

	Circuit*		
	Fuel	Oxidizer Secondary Combustor	Oxidizer Primary Combustor
Pump Discharge	2000	1740	1740
ΔP, Line	35	20	35
ΔP, Pump Discharge Valve	35	50	35
Coolant Jacket Inlet	1930	----	----
ΔP, Coolant Jacket	250	----	----
Primary Combustor Valve Inlet	1680	----	1670
ΔP, Primary Combustor Valve	35	----	25
Primary Combustor Injector Inlet	1645	----	1645
ΔP, Primary Combustor Injector	50	----	50
Primary Combustor Injector Face	1595	----	1595
Primary Combustor Plenum	1545	----	1545
Pressure Ratio, Primary Combustor to Turbine Nozzle	1.017	----	1.017
ΔP, Primary Combustor Nozzle	25	----	25
Turbine Nozzle Inlet	1520	----	1520
Turbine Pressure Ratio	1.117	----	1.117
ΔP, Turbine	158	----	158
Turbine Exit	1362	----	1362
Orifice Inlet	----	1670	----
ΔP, Orifice and Valve	----	248	----
Secondary Combustor Injector Inlet	1362	1422	1362
ΔP, Secondary Combustor Injector	60	120	60
Secondary Combustor Injector Face	1302	1302	1302
Secondary Plenum	1265	1265	1265

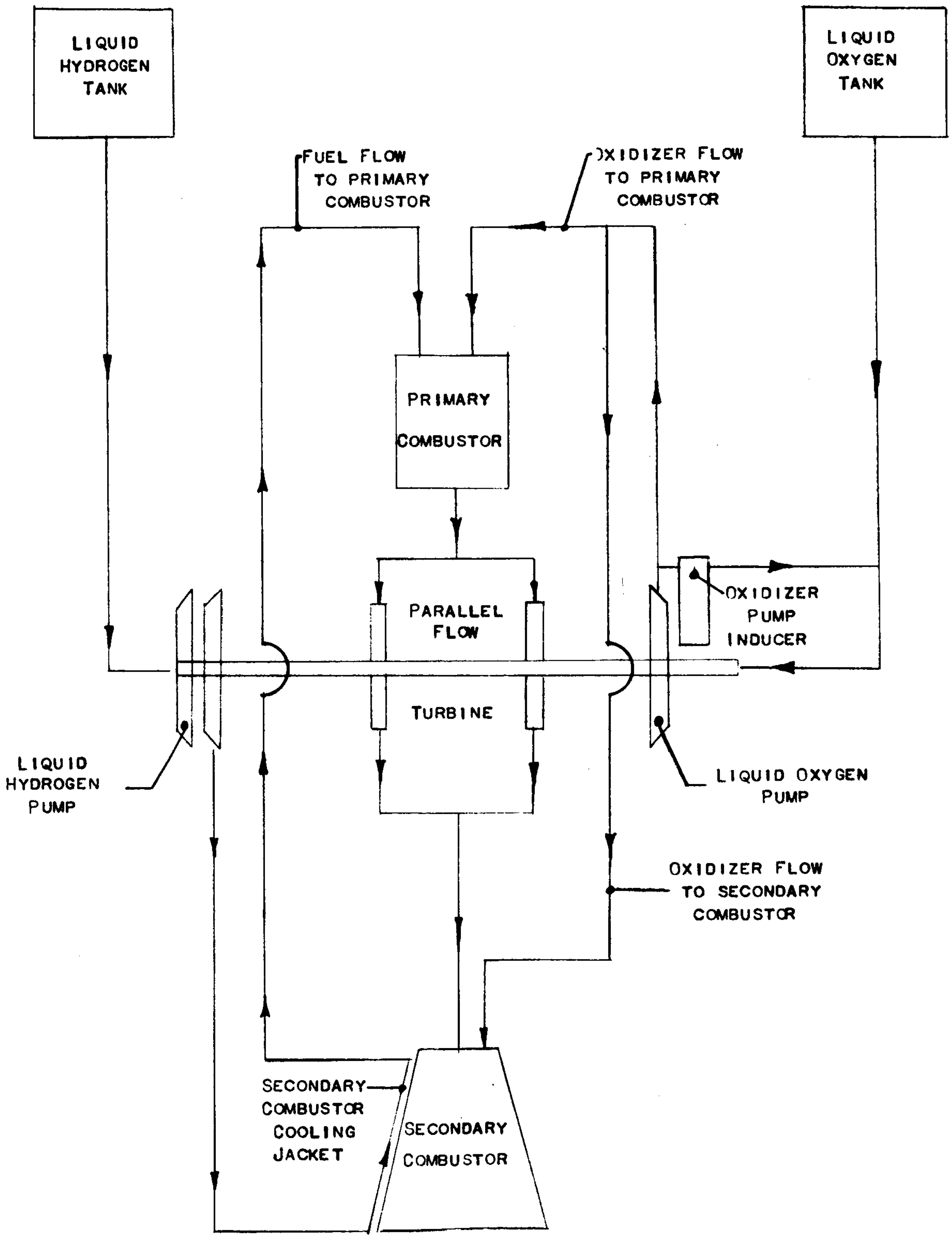
*Pressures are in psia; pressure drops are in psi.

TABLE VI-A-9

AJ-6000 ENGINE PRESSURE SCHEDULE

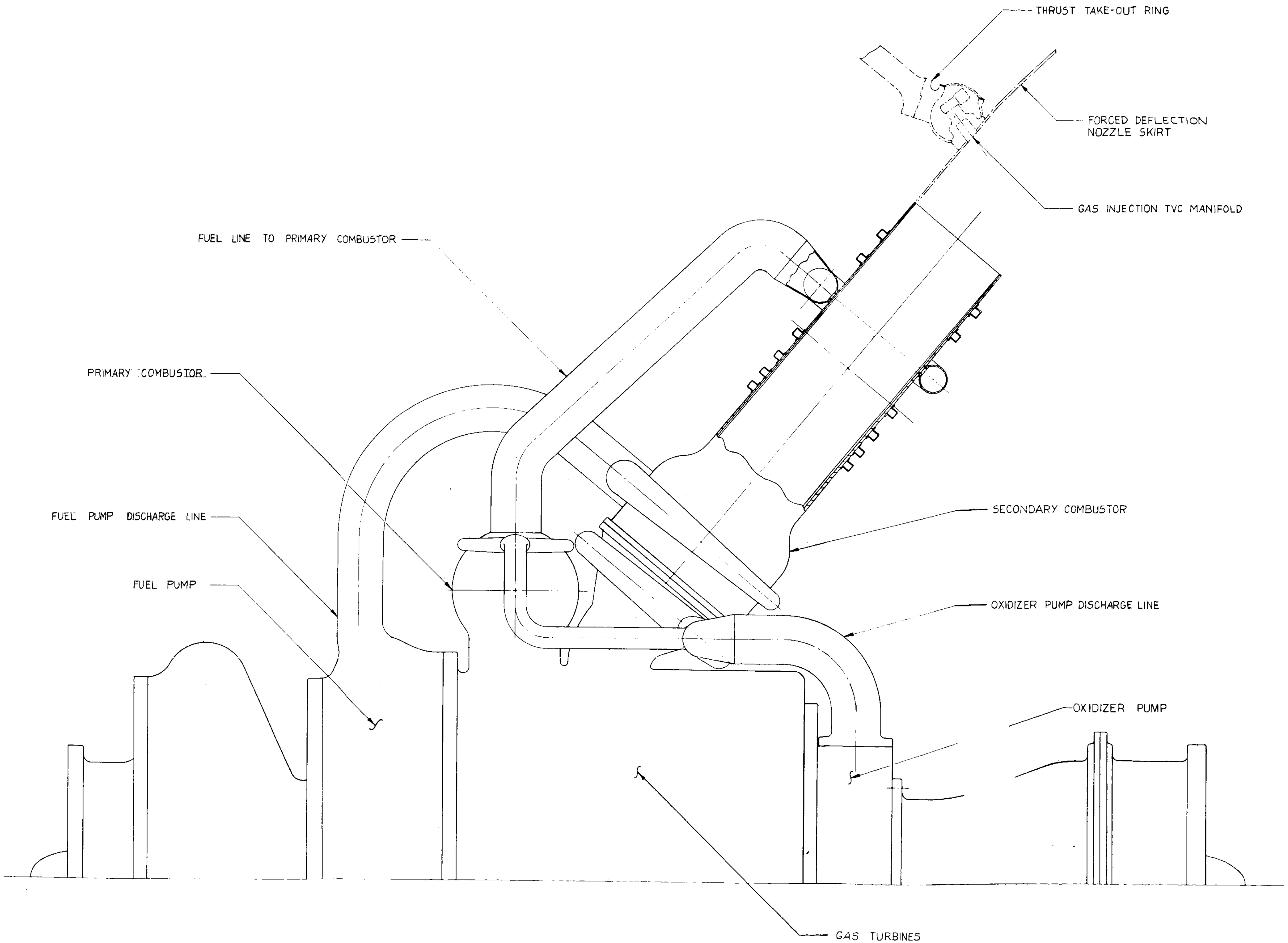
	Circuit*		
	Fuel	Oxidizer Secondary Combustor	Oxidizer Primary Combustor
Pump Discharge	6000	5375	5375
ΔP, Line	50	50	50
ΔP, Pump Discharge Valve	50	50	50
Coolant Jacket Inlet	5900	----	----
ΔP, Coolant Jacket	500	----	----
Primary Combustor Valve Inlet	5400	----	5275
ΔP, Primary Combustor Valve	100	----	75
ΔP, Primary Combustor Injector	350	----	250
Primary Combustor Injector Face	4950	----	4950
Primary Combustor Plenum	4810	----	4810
Pressure Ratio, Primary Combustor to Turbine Nozzle	1.027	----	1.027
ΔP, Primary Combustor to Turbine Nozzle	125	----	125
Turbine Nozzle Inlet	4685	----	4685
Turbine Pressure Ratio	1.399	----	1.399
ΔP, Turbine	1335	----	1335
Turbine Exit	3350	----	3350
Orifice Inlet	----	5275	----
ΔP, Orifice and Valve	----	1850	----
ΔP, Turbine Exit to Secondary Combustor Injector	25	----	25
Secondary Combustor Injector Inlet	3325	3425	3325
ΔP, Secondary Combustor Injector	150	250	150
Secondary Combustor Injector Face	3175	3175	3175
Secondary Combustor Plenum	3085	3085	3085

*Pressures are in psia; pressure drops are in psi.



AJ-1 Engine Flow Schematic

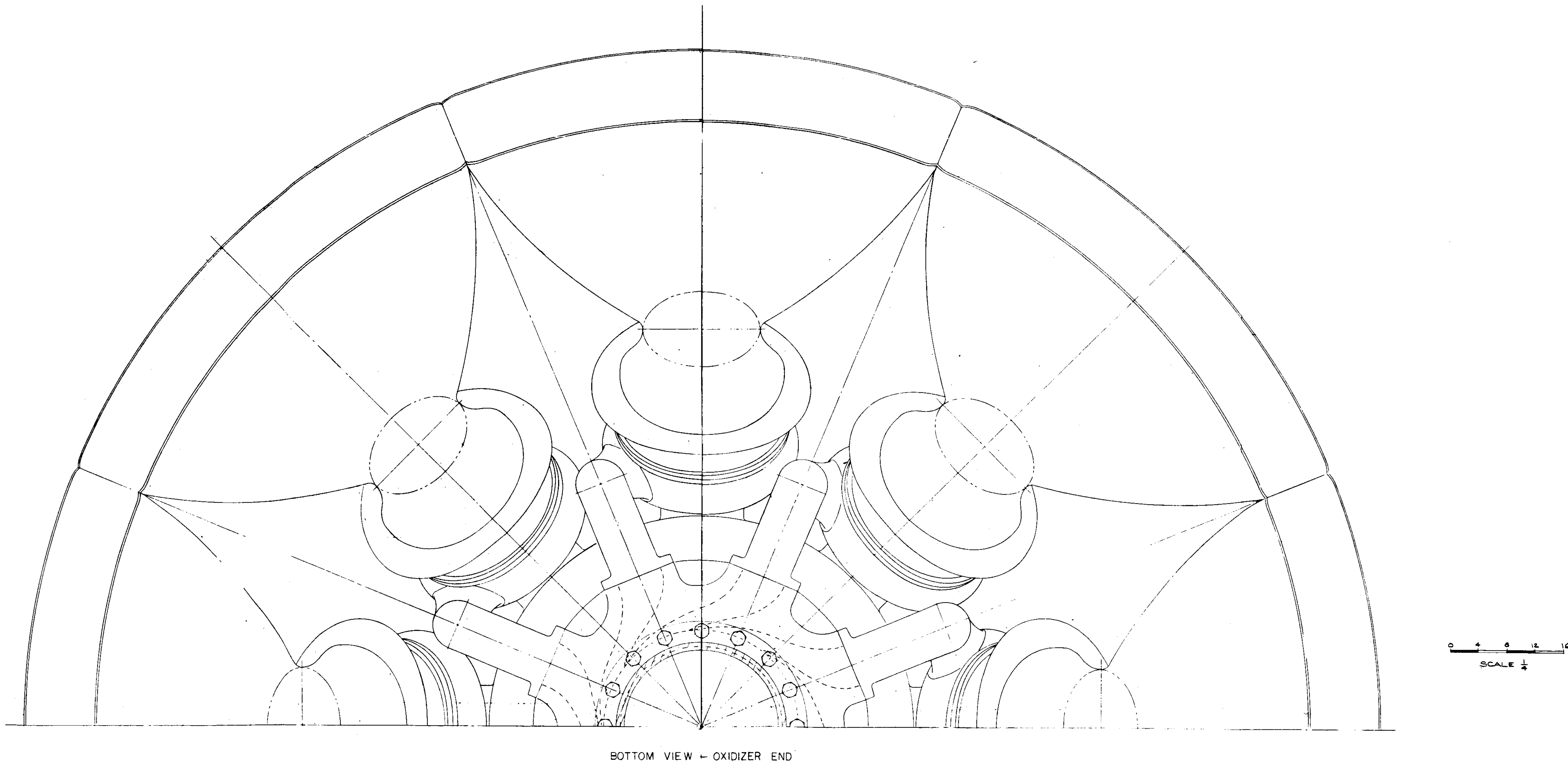
Figure VI-A-1



SEA LEVEL THRUST LBS	6,000,000
PROPELLANTS	LOX/LH_2
NOZZLE AREA RATIO	120
MIXTURE RATIO-THRUST CHAMBER	6.0
MIXTURE RATIO-PRIMARY COMBUSTOR	0.915
THRUST CHAMBER PRESSURE PSIA	2500

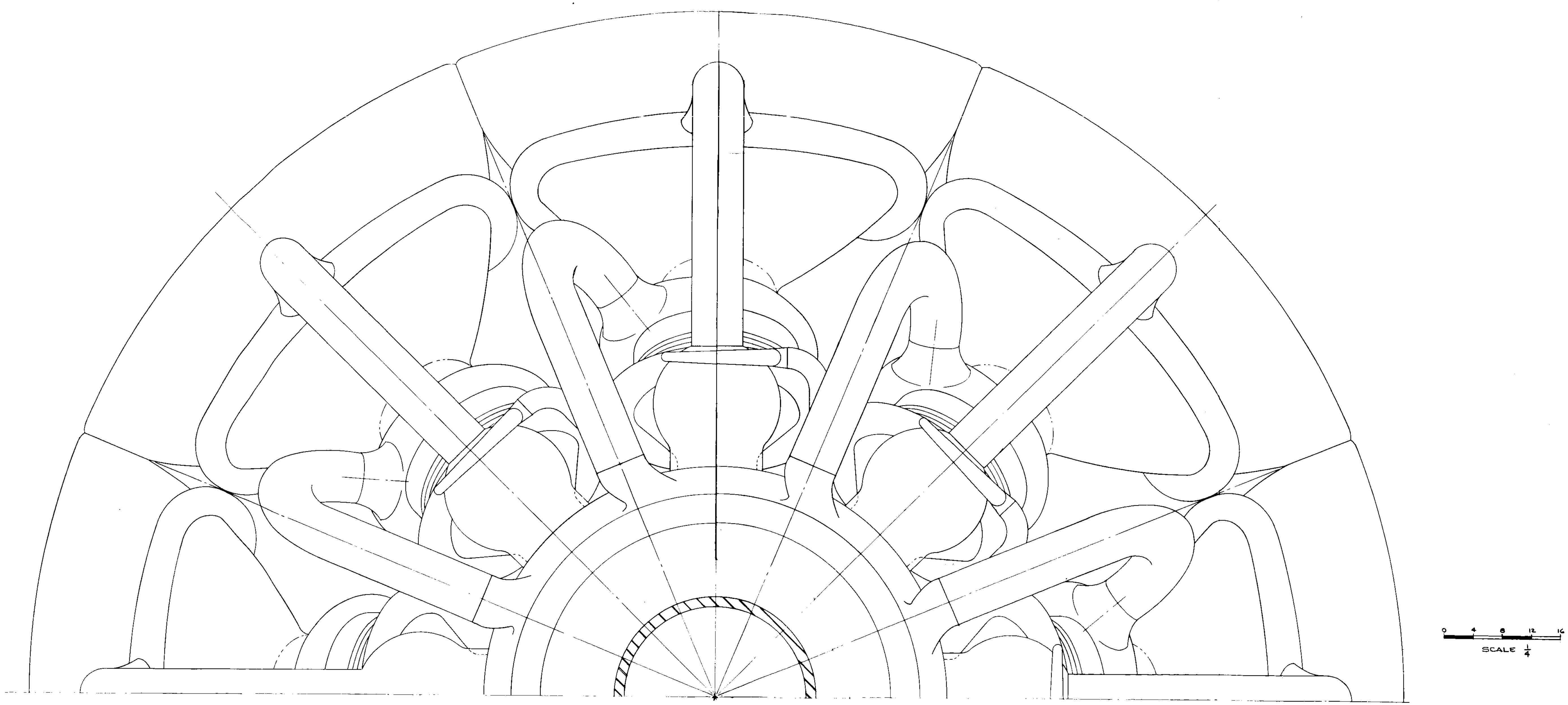
AJ-1 Engine, Vertical Section

Figure VI-A-2



AJ-1 Engine, Bottom View

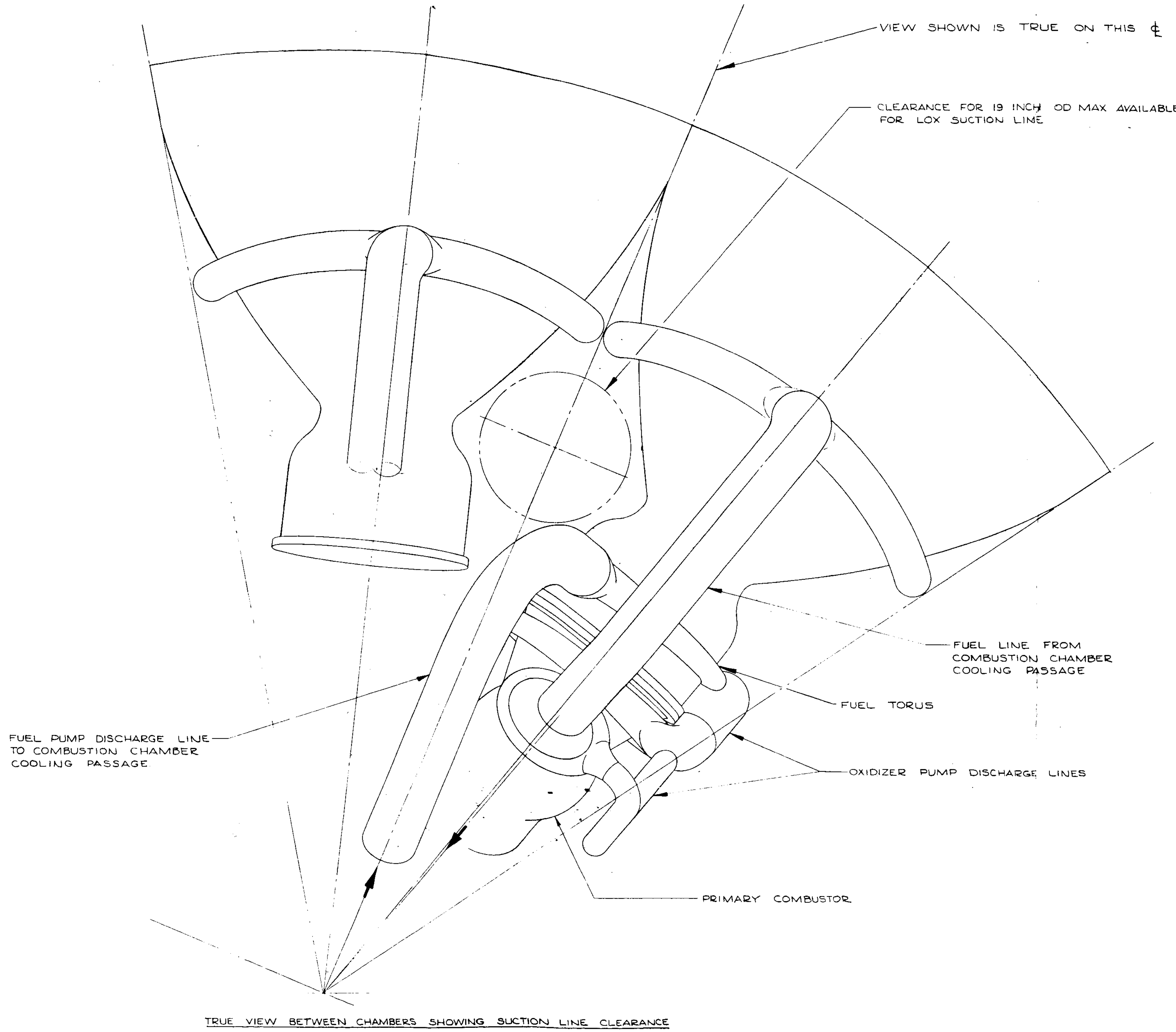
Figure VI-A-3



TOP VIEW - FUEL PUMP END

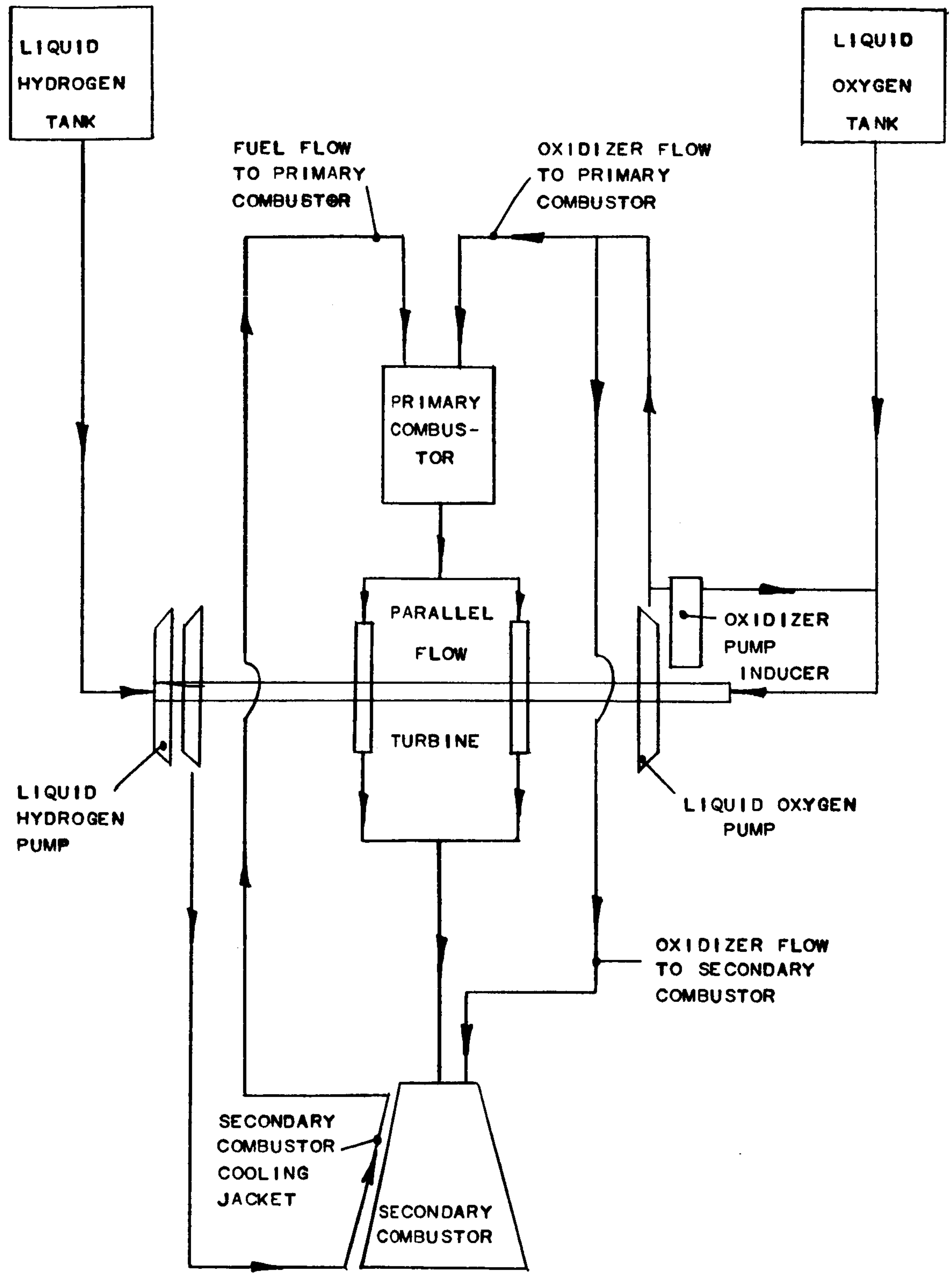
AJ-1 Engine, Top View

Figure VI-A-4



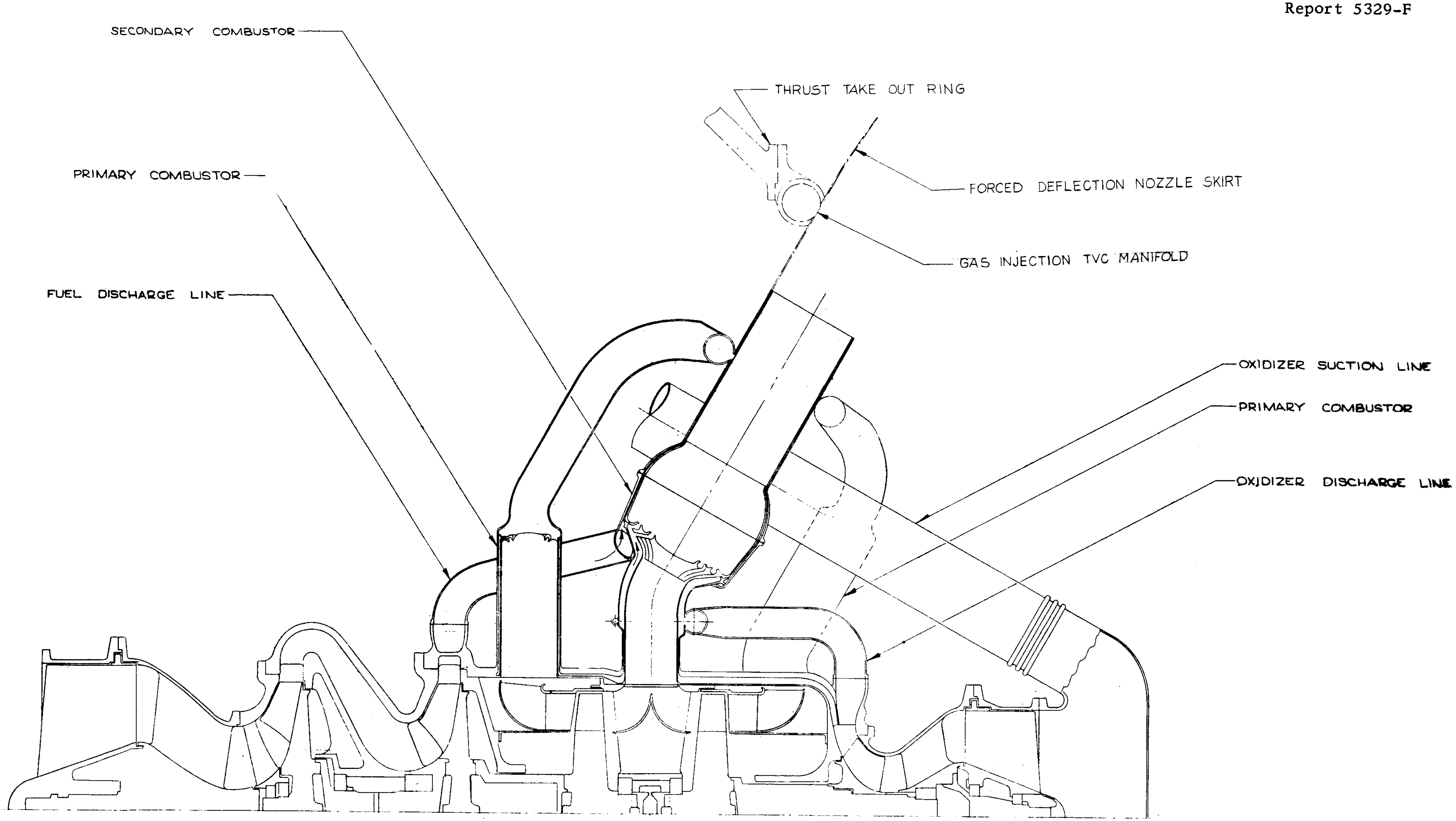
AJ-1 Engine, Suction Line Clearance

Figure VI-A-5

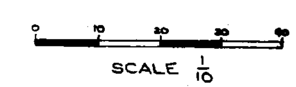


AJ-2A Engine Flow Schematic

Figure VI-A-6

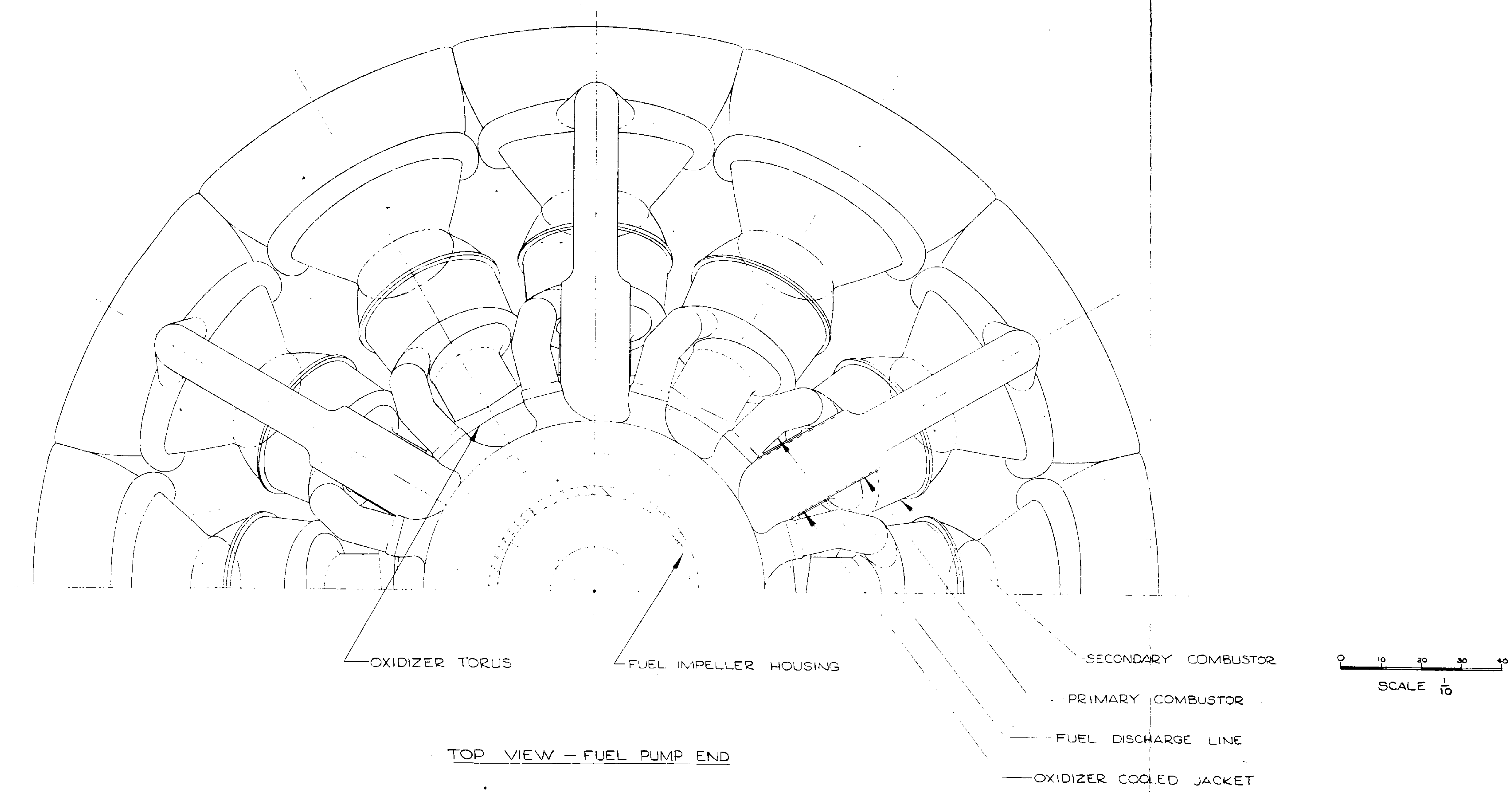


SEA LEVEL THRUST LBS	24,000,000
PROPELLANTS	LOX/LH_2
NOZZLE AREA RATIO	120
MIXTURE RATIO - THRUST CHAMBER	6.0
MIXTURE RATIO - PRIMARY COMBUSTOR	0.915
THRUST CHAMBER PRESSURE, PSIA	2500



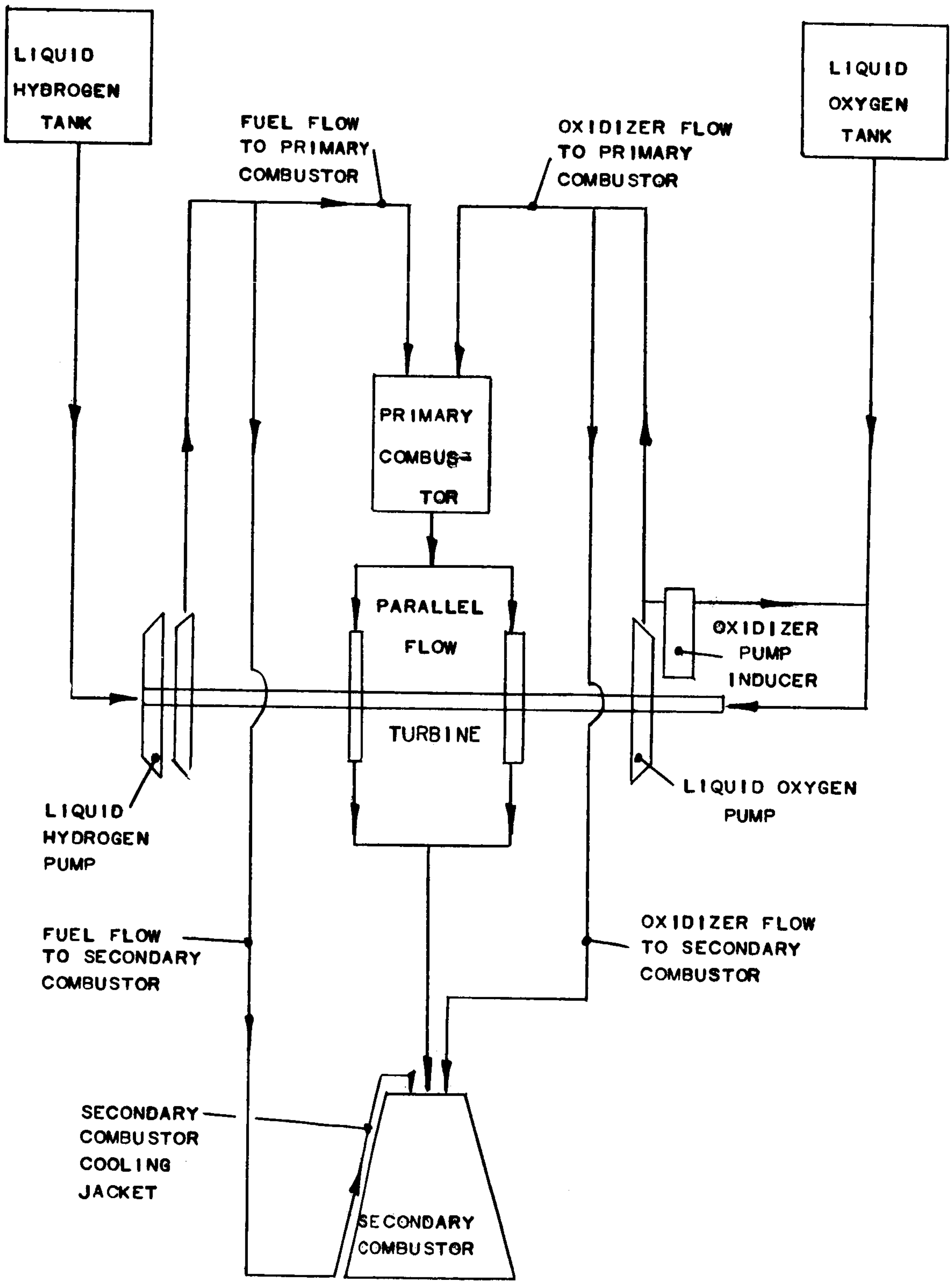
AJ-2A Engine, Vertical Section

Figure VI-A-7



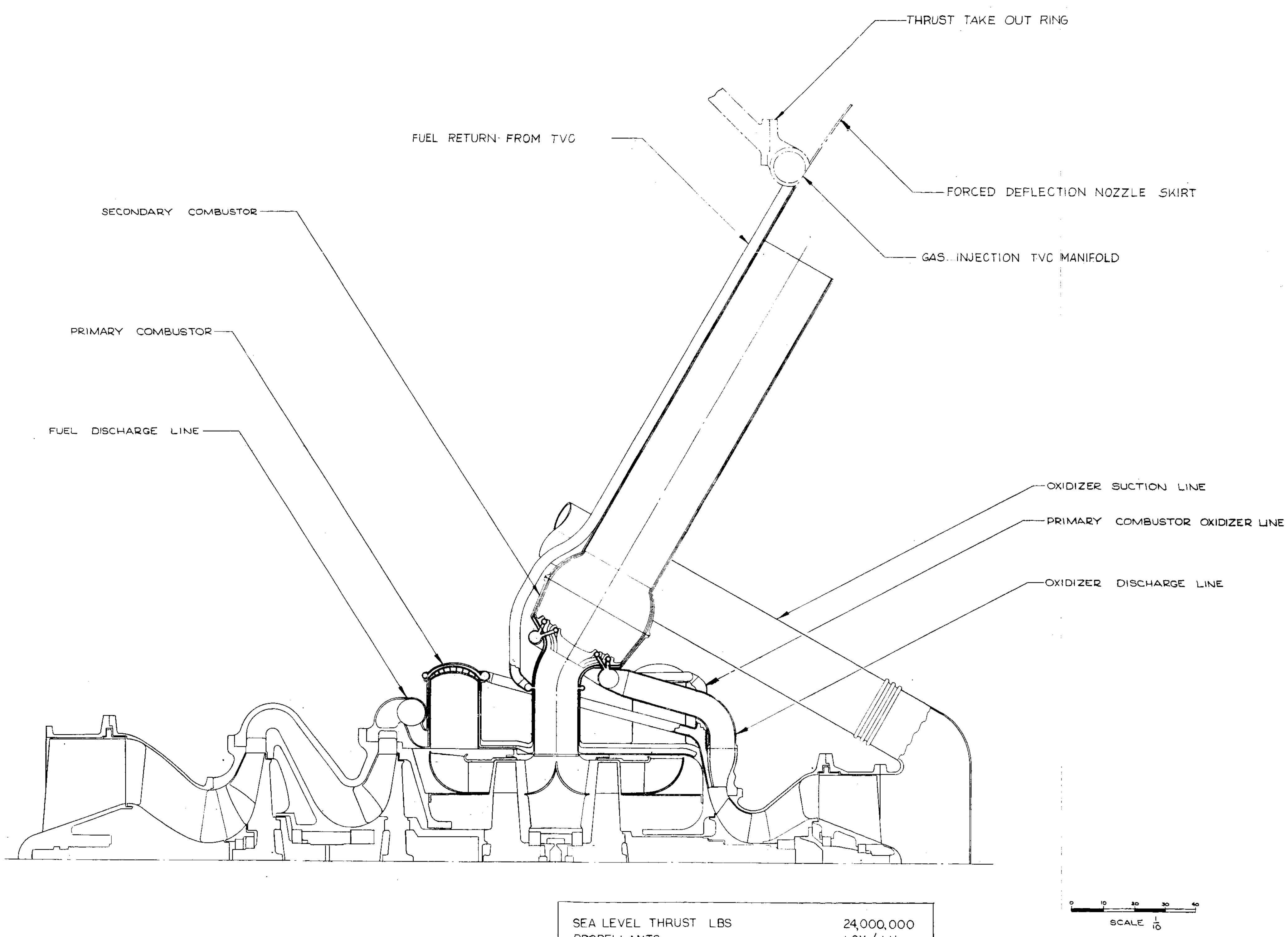
AJ-2A Engine, Top View

Figure VI-A-8



AJ-2B Engine Flow Schematic

Figure VI-A-9



SEA LEVEL THRUST LBS	24,000,000
PROPELLANTS	LOX / LH_2
NOZZLE AREA RATIO	120
MIXTURE RATIO - THRUST CHAMBER	6.0
MIXTURE RATIO - PRIMARY COMBUSTOR	0.915
THRUST CHAMBER PRESSURE PSIA	2500

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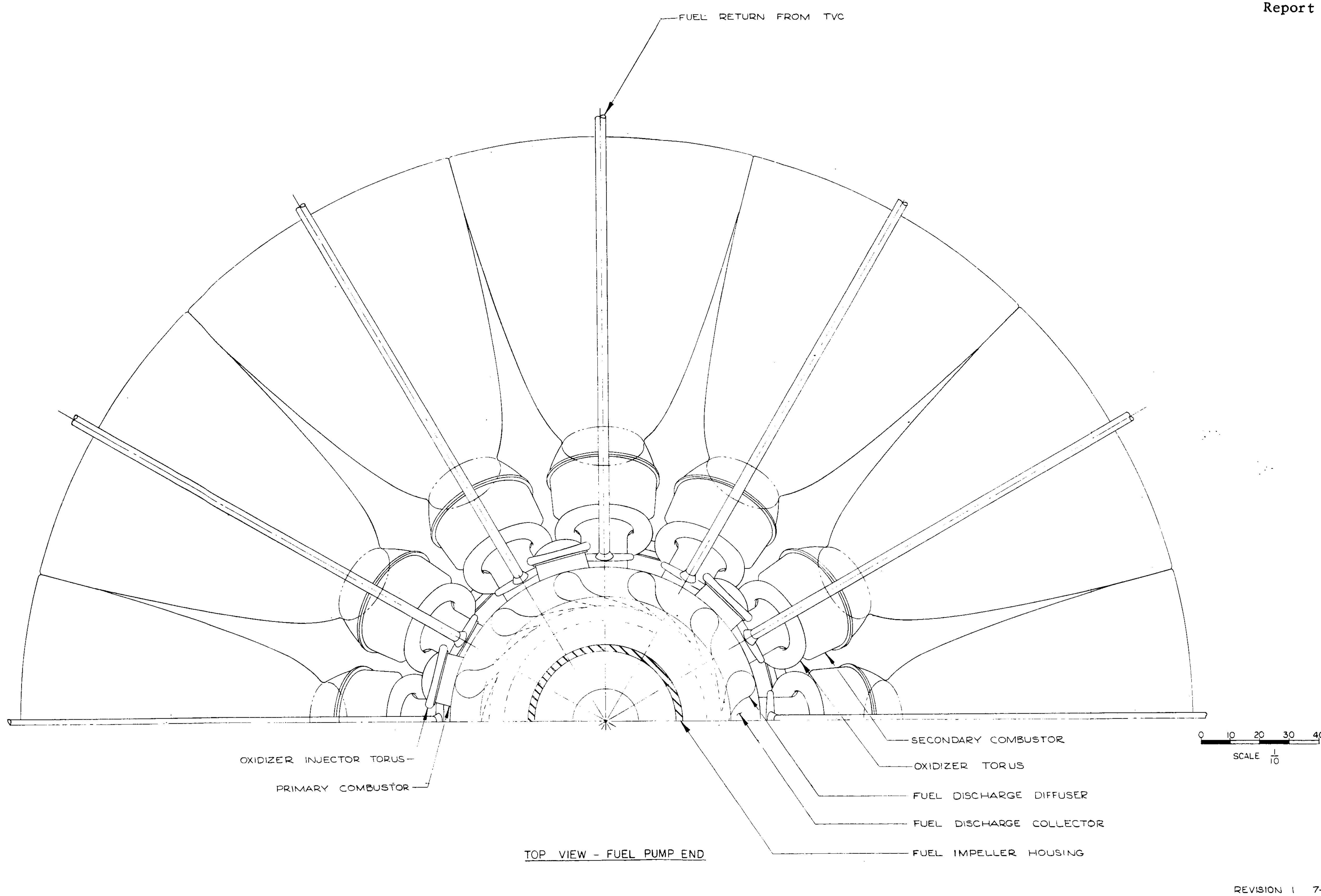
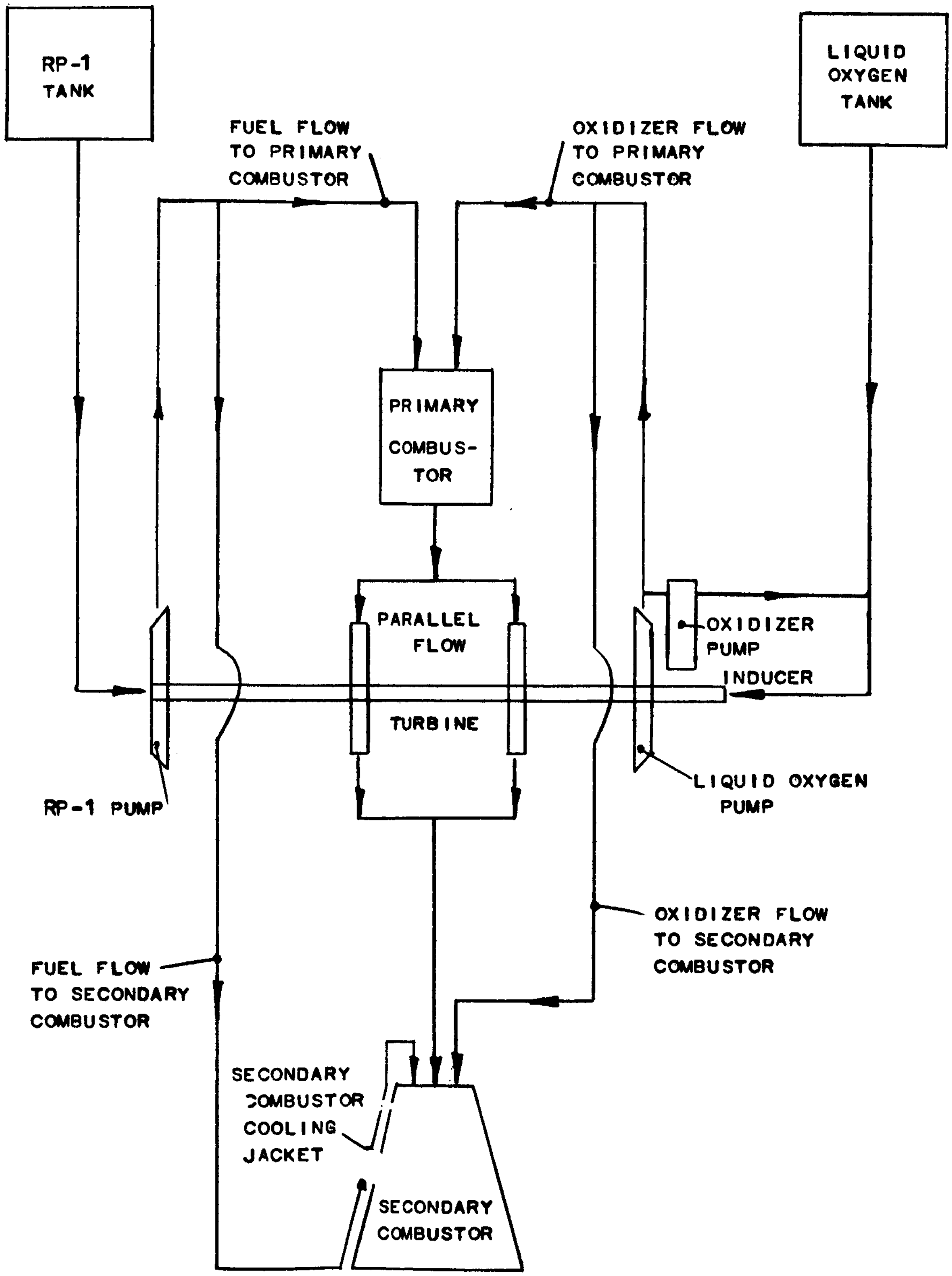
AJ-2B Engine, Vertical Section

Figure VI-A-10



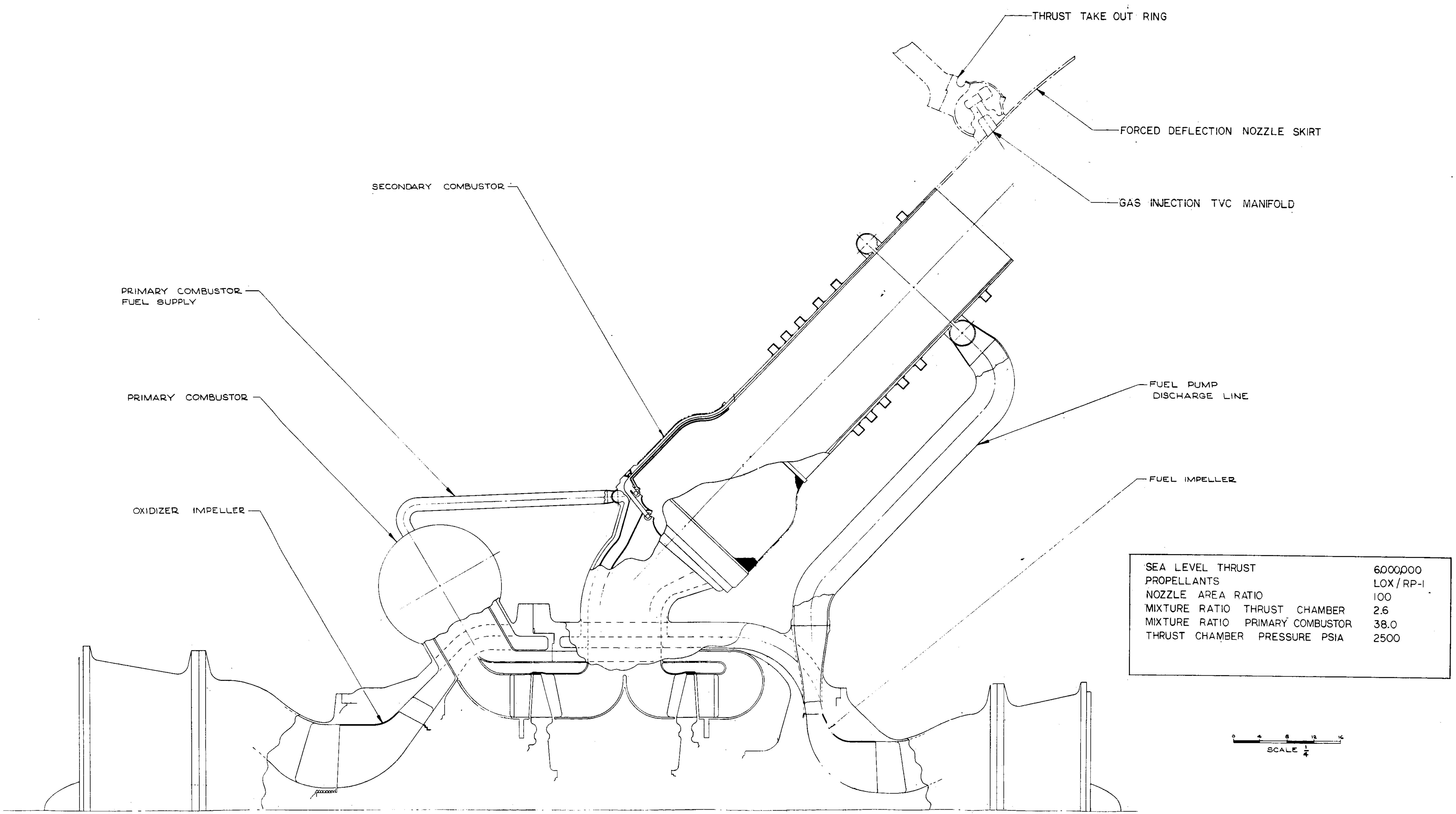
AJ-2B Engine, Top View

Figure VI-A-11



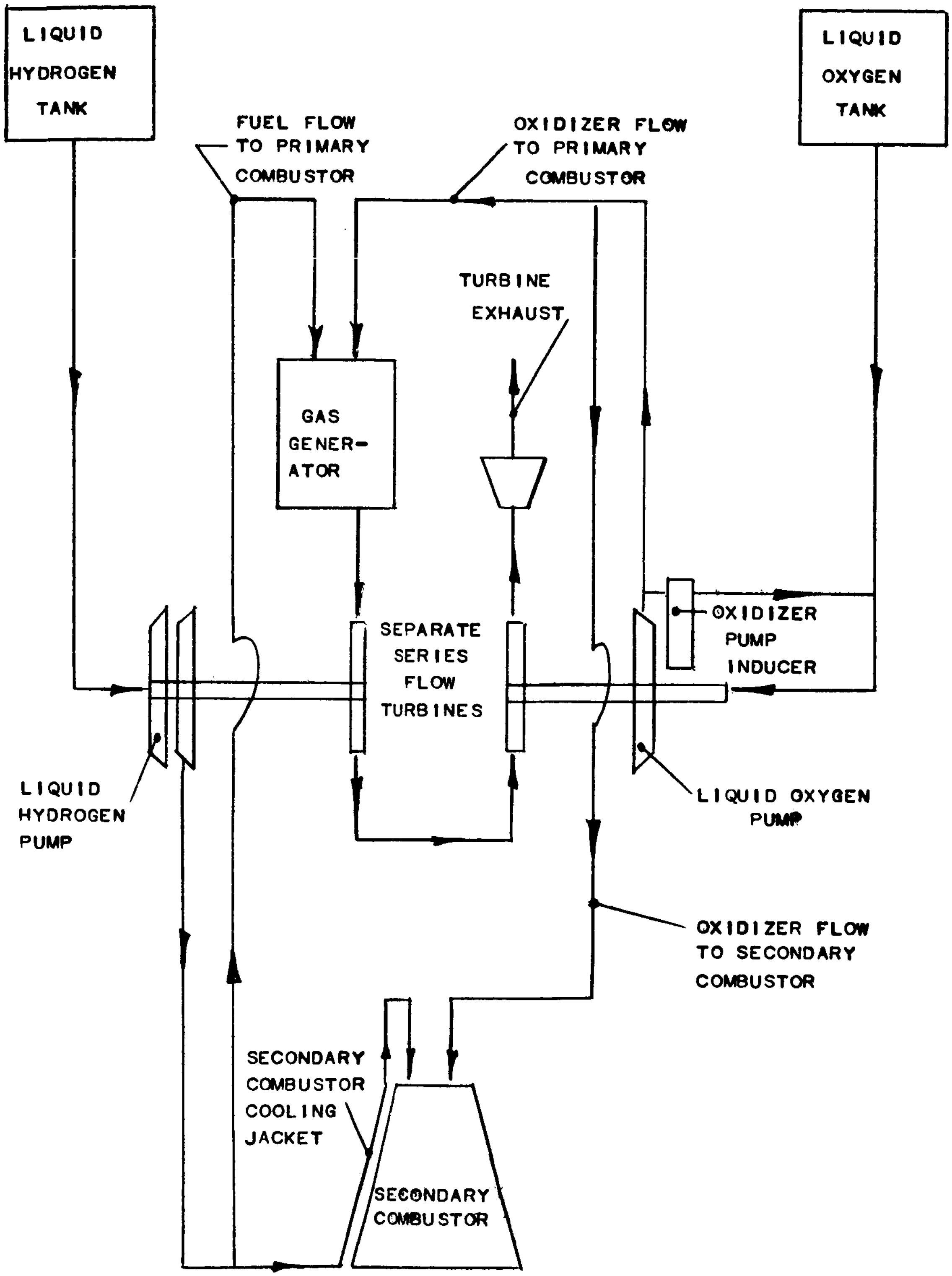
AJ-3 Engine Flow Schematic

Figure VI-A-12



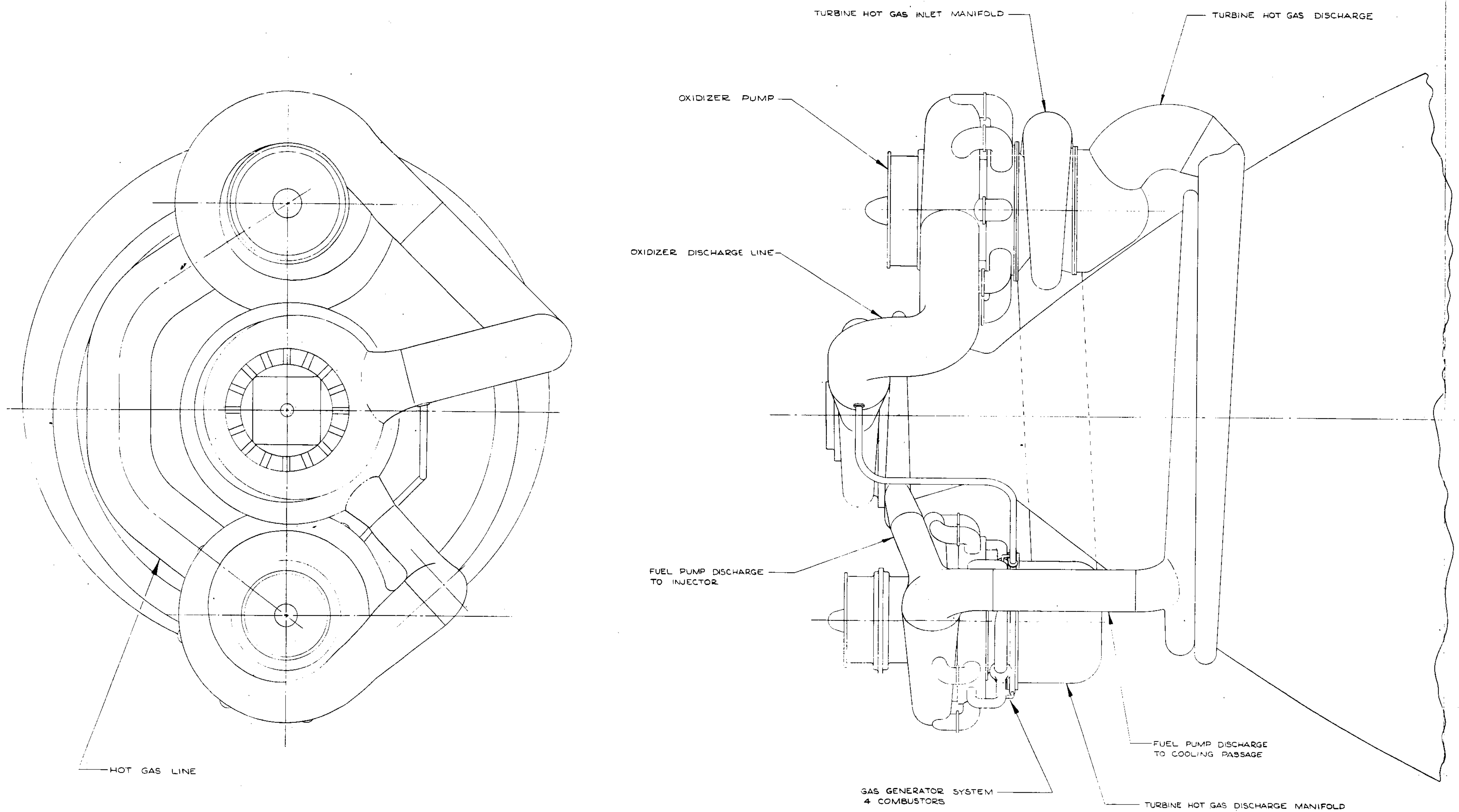
AJ-3 Engine, Vertical Section

Figure VI-A-13

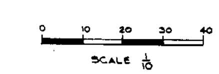


AJ-5 Engine Flow Schematic

Figure VI-A-14



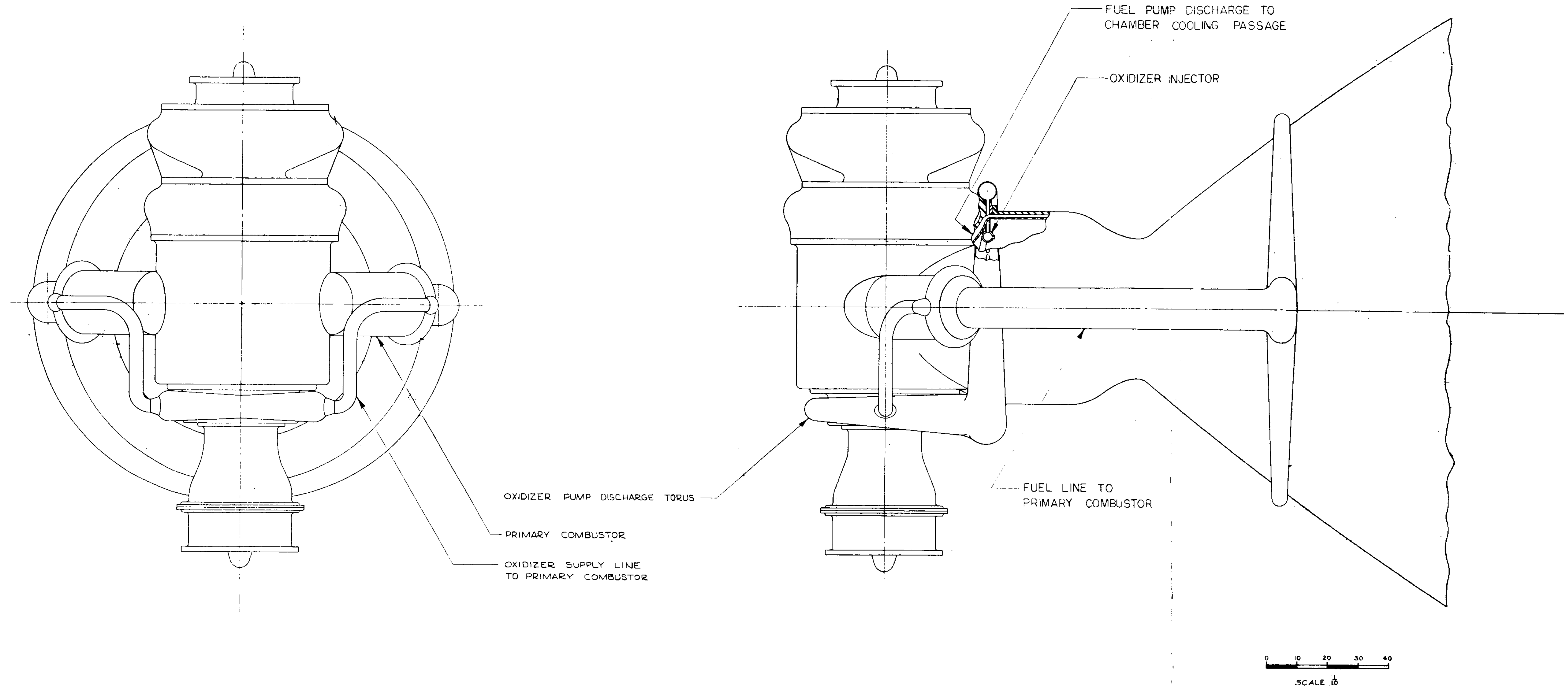
SEA LEVEL THRUST LBS	6,000,000
PROPELLANTS	LOX/LH_2
NOZZLE AREA RATIO	50
MIXTURE RATIO - THRUST CHAMBER	6.0
MIXTURE RATIO - GAS GENERATOR	0.915
THRUST CHAMBER PRESSURE PSIA	3400



AJ-5 Engine

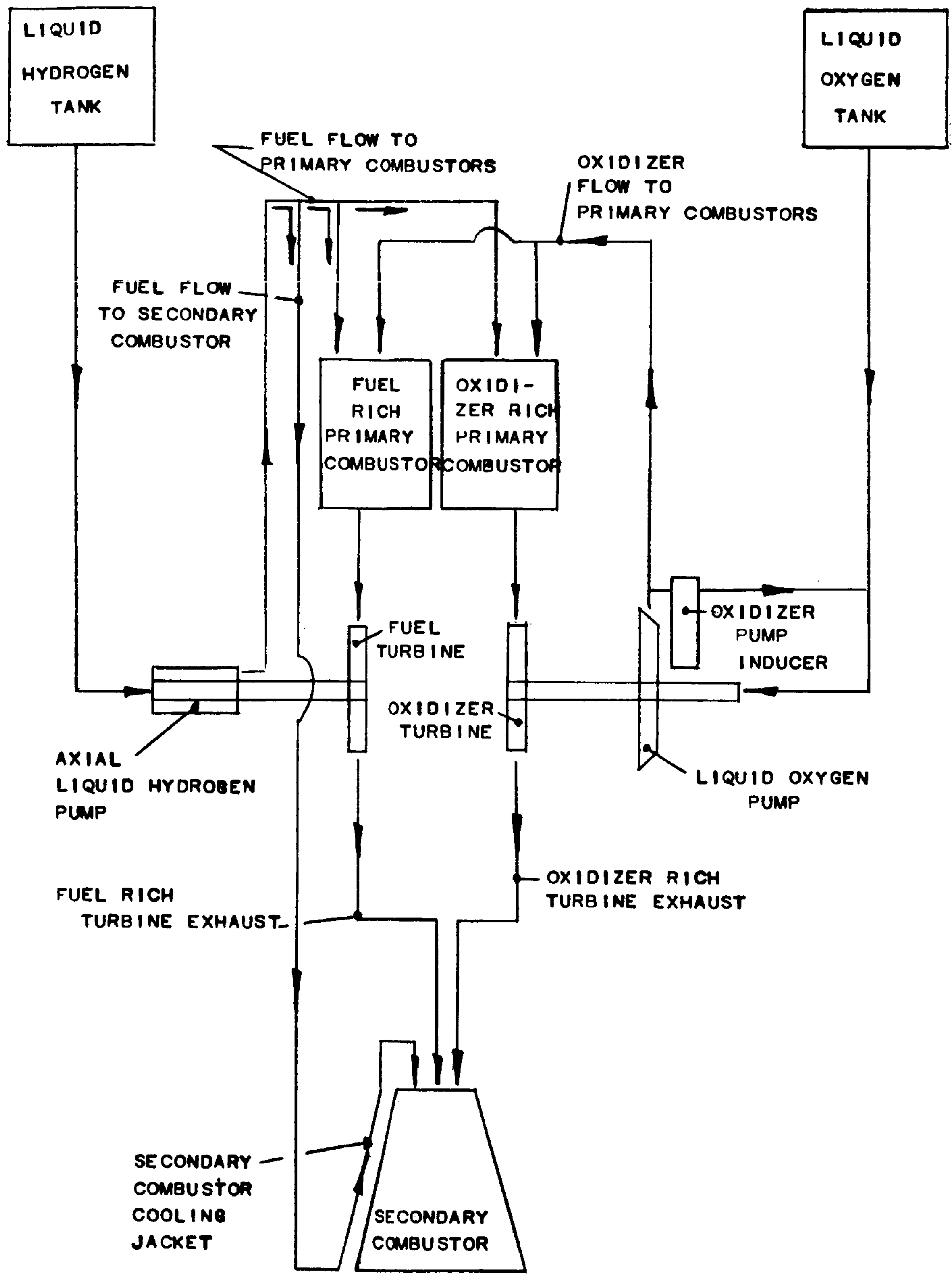
Figure VI-A-15

SEA LEVEL THRUST LBS	6,000,000
PROPELLANTS	LOX/LH$_2$
NOZZLE AREA RATIO	40
MIXTURE RATIO THRUST CHAMBER	6.0
MIXTURE RATIO PRIMARY COMBUSTOR	0.915
THRUST CHAMBER PRESSURE PSIA	2500



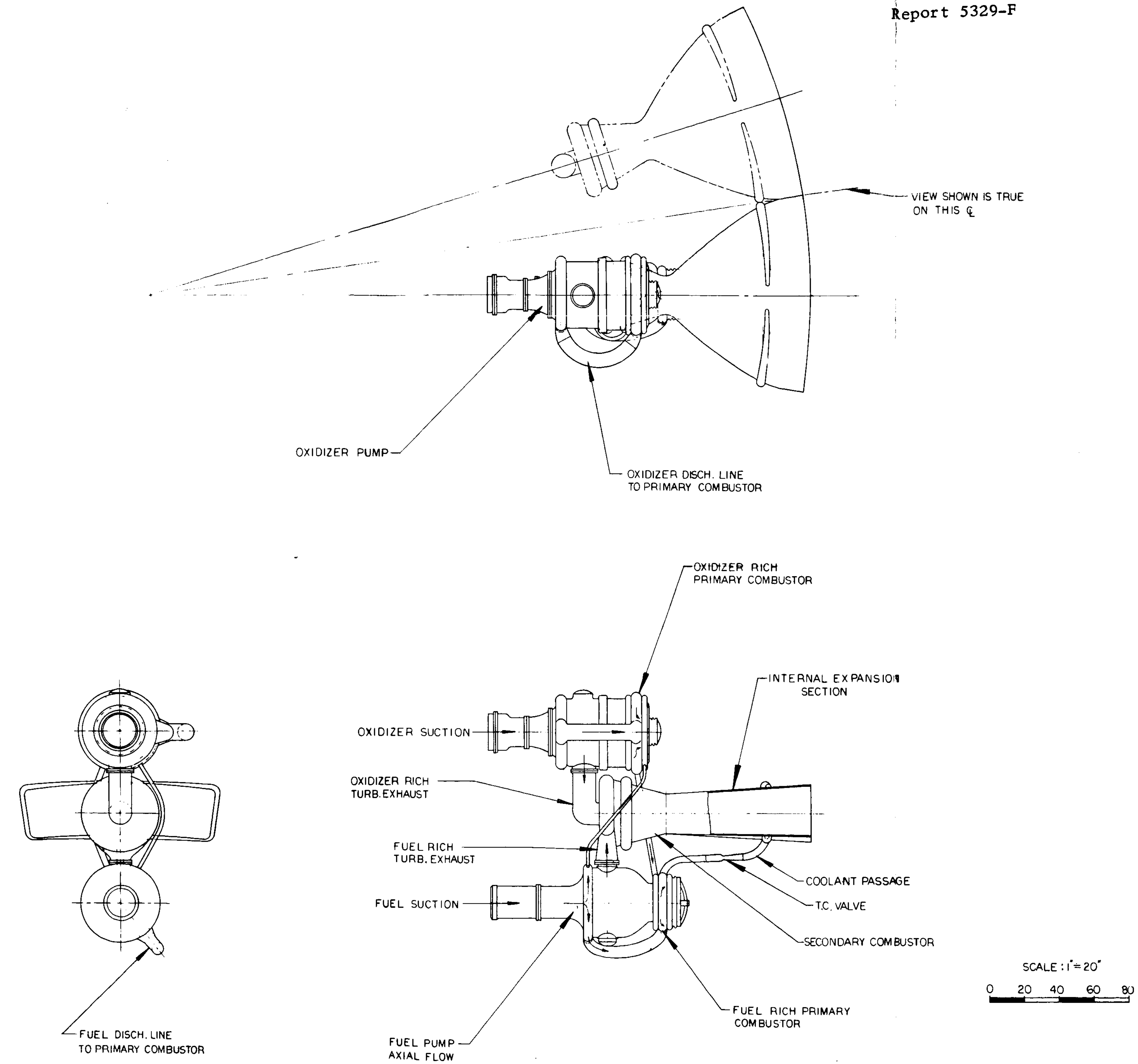
AJ-6 Engine

Figure VI-A-16



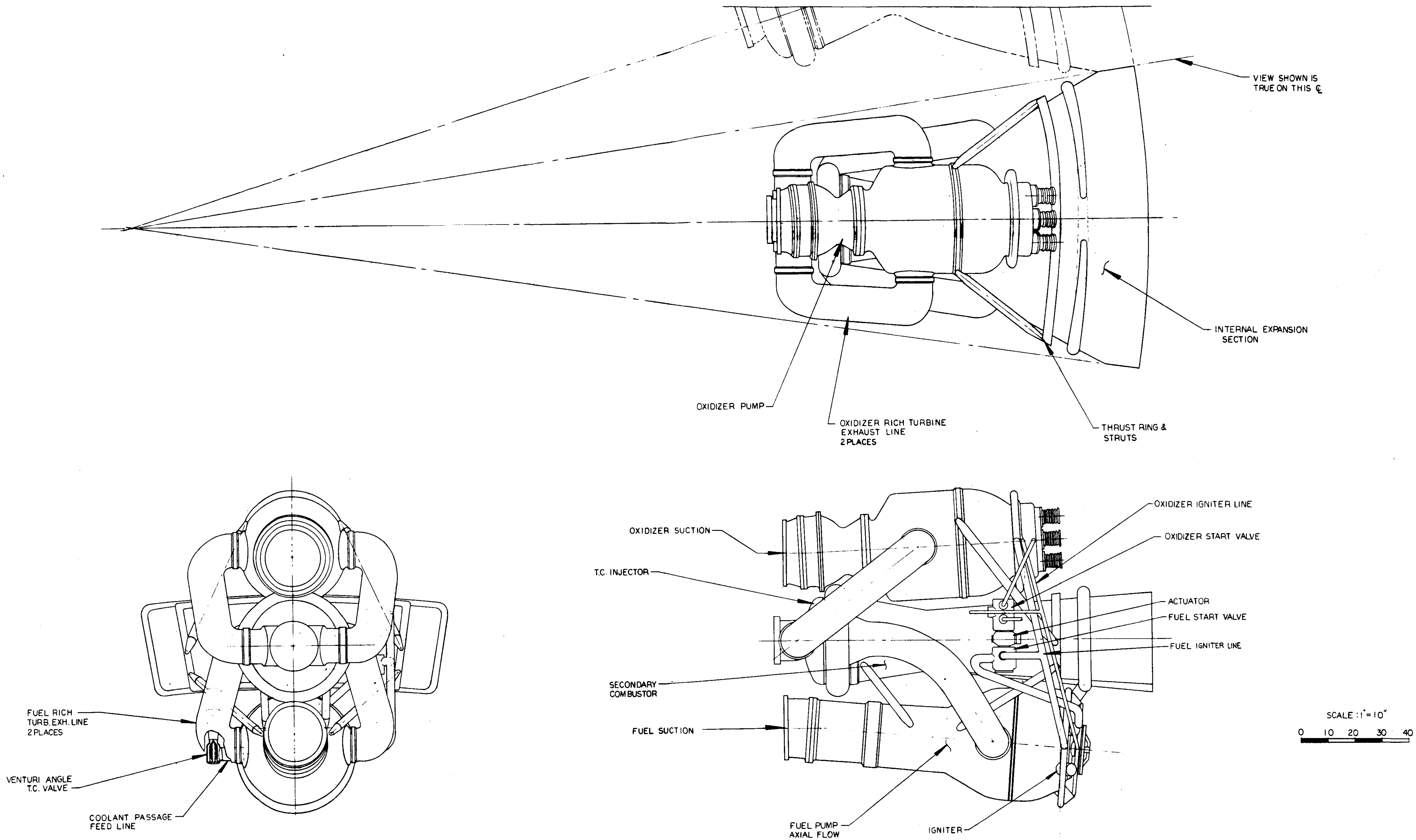
AJ-8 Engine Flow Schematic

Figure VI-A-17



AJ-8A Engine

Figure VI-A-18



AJ-8B Engine

Figure VI-A-19

VI, Engine-Systems Studies (cont.)

B. STEADY-STATE ENGINE-SYSTEM ANALYSIS

1. Design-Point Analysis

The purpose of the design-point analysis was to firmly establish a balance point (which could be used as a basis for all later studies) for each engine, including off-design analysis, auxiliary-systems studies, and transient analysis. Because the calculations necessary to perform these later studies involve a great number of equations and interactions between equations, a high-speed, digital computer is necessary to perform these calculations. An IBM 7094 Computer was used to accomplish this task. The program used simulates pump-fed, bipropellant engines.

The liquid engine design computer program solves a series of equations in each component area (i.e., thrust chamber, main oxidizer line, fuel pump) and then continues on to the next component. An initial estimate is programmed in and a calculation is made. The result is compared to the original estimate and a second estimate is made. This process is repeated until any difference is resolved. Variables for the various equations come from separate input values, previously calculated values, or curves. A sample operation is described below, using pump head as an example.

$$H = H_D \left(\frac{N}{N_D} \right)^2 \left(C_{34} + C_{35}\,\phi + C_{36}\,\phi^2 + \ldots + C_{41}\,\phi^7 \right)$$

where H = pump head, ft

H_D = pump design head, ft

N = pump speed, rpm

N_D = pump design speed, rpm

C_i = constants from curve fit

ϕ = $(Q/N)/(Q/N)_D$

where Q = pump volumetric flow, ft^3/sec

VI, B, Steady-State Engine-System Analysis (cont.)

The values for H_D and N_D are input data, N is established by previous turbine calculations, and the C values are the coefficients of a polynominal curve fit of the pump characteristics (head versus capacity and torque versus capacity).

The input for this program, as mentioned above, consists of the basic engine specifications and pressure schedules as summarized in the tables of Section VI, A. Line, valve, and injector resistances are determined from the specified pressure drops at design flow rates. Because these resistances remain constant, pressure drop becomes a function of flow rate and density. Finally, performance curves for the turbomachinery, represented by polynomial expressions, are placed in the program. These curves enable the program to balance the engine operation at off-design conditions.

2. Off-Design Analysis

a. Statistical-Component Tolerance Study

The objective of the statistical off-design analysis is to establish a range of off-design pressures that might realistically be expected in a random group of assembled engines, the off-design values being attributed to component and engine-balance tolerances.

The analysis utilized a variation of the steady-state program discussed in the previous section. Certain tolerance levels which can realistically be expected have been established on the basis of component-performance tests and engine-assembly firings of Titan hardware. These observed component tolerances are caused by manufacturing variations and result in the off-design engine performance. A summary of these tolerances is shown in Table VI-B-1. The statistical

VI, B, Steady-State Engine-System Analysis (cont.)

program assumes an infinite collection of values for each specification considered so that the distribution is normal and the three-sigma variation (99.73% of all values) is equal to the established tolerance on that specification. The program randomly selects up to 50 values for each specification and, in effect, assembles and balances the resultant random engines. The program then balances the simulated engine in the same manner as a hardware engine would be balanced. The suction conditions are set at design values. MR_{TC}, MR_{PC}, and F* are measured and usually found to be slightly off-design. The balance orifices (in the oxidizer-discharge and primary combustor lines for AJ-1) are adjusted until the mixture ratios reach these design values (6.0 for the thrust chamber and 0.915 for the primary combustor for AJ-1). Thrust will remain slightly off-design because of variation within tolerance of other engine specifications (i.e., pump head or thrust-chamber throat area). Its value is then tested to determine if it falls within the acceptable tolerance range of ±3%. Neither a small sample of values from a very large collection nor an algebraic function of these samples will fit a normal distribution curve. However, the assumption is made, for the purpose of analyzing these results, that thrust and pump discharge pressure will fit a normal distribution, and the straight line of best fit is drawn, on probability paper, through 25 calculated points (25 sample engines were included in the analysis of each engine).

Figure VI-B-1 shows partial results of the study performed for Engines AJ-1, AJ-2A, AJ-3 and AJ-5. The horizontal dashed lines indicate the acceptable thrust tolerance of ±3% (based on Titan experience). The figure shows that between 21 and 36% of the analytical assemblies fail to reach the minimum acceptable thrust. This can be attributed to the fact that the basic engines were designed without the balancing orifices described above. That is, there is no capability for opening orifices and reducing pressure drops if the engine components fail to reach the minimum specification. The results indicate that for 99.9% of all engines to reach at least minimum acceptable thrust, an over-design factor of 11% of rated thrust must be used. This value is determined by measuring the difference between the 97% minimum acceptable thrust and the 86% of thrust achieved by 99.9% of the worst engine samples. A study of existing engine systems, such as M-1, indicates that this 11% value is not unreasonable.

*MR_{TC} = thrust chamber (secondary combustor) mixture ratio; MR_{PC} = primary combustor (gas generator) mixture ratio; F = engine thrust, lb

VI, B, Steady-State Engine-System Analysis (cont.)

The primary conclusion to be drawn from this study is that variations in the basic engine parameters of thrust, pressure, cycle, and propellant have little effect on the engine sensitivity to component tolerances. Examination of Table VI-B-1 will show that, of the component tolerances listed, none are pressure-dependent functions.

Figures VI-B-2 and -3 show the statistical variation of fuel- and oxidizer-pump discharge pressures (P_D). As in the case of the thrust results, any variation between the engines is due to the random nature of the study rather than any significant difference in characteristics due to changes in size, pressure, cycle, or propellant. In a 25-engine sample, regardless of engine type, the maximum variation of pump discharge pressures can be expected to be $\pm 10\%$ where the allowable tolerance on pump head was $\pm 3\%$. This tolerance is on head at a given speed and capacity. The additional variation in P_D is caused by off-design operation in the various other components (i.e., pressure drop, flow rate, etc.). These curves show the same characteristic as the thrust variation because a large number (20%) of the pumps fail to meet their minimum specification. The two results are interdependent, and the thrust variation is primarily a function of the discharge-pressure variation. The thrust condition can be corrected by increasing the pump discharge pressures at the design point by 11%, as indicated by Figures VI-B-2 and -3. This increase results in 99.9% of the engines meeting the minimum specification of -5% of pump discharge pressure. It can be concluded from this study that there is a minimal effect on the percentage of over pressure necessary because of changes in cycle, thrust, or propellant.

b. Vehicle-Acceleration Effects

All nominal engine designs on this program assume a 1 g longitudinal loading on the vehicles, i.e., full-thrust operation during vehicle hold-down. The vehicle loading then rises from 1.25 g at liftoff to 6 or 10 g at burnout or throttle back (6 g for man-rated vehicle or 10 g for unmanned). It was desired in this study to determine the severity of the problems created within the engine system and the auxiliary components by this high longitudinal loading.

VI, B, Steady-State Engine-System Analysis (cont.)

The most prominent result of this loading is to effectively increase the static head of the propellant above the pump inlet. Using the computer program, several cases with different longitudinal loadings were calculated for the AJ-1 engine. With the LO_2 tank on top, the major off-design conditions occur in the oxidizer system. The net positive suction head (NPSH) is greatly increased. This results in a series of inter-related effects: (1) The oxidizer flow rate increases. (2) The primary combustor mixture ratio and energy to the turbine increases. (3) The turbopump speed increases. (4) The pump head increases. The total increase in pump discharge pressure resulting from all of these effects is as much as 27%, while other engine operating conditions, such as turbine-inlet temperature, reach intolerable values. These effects emphasize the need for some form of pressure control in this and any large engine.

Comparative pressures throughout the system and significant engine specifications for 1, 1.25, 6, and 10 g are shown in Table VI-B-2 and Figures VI-B-4, -5 and -6.

The effects of this acceleration on the design of the oxidizer suction (feed) lines as well as the high-pressure ducting within the engine are discussed in Section VIII, B.

3. <u>Associated System Requirements and Effects</u>

a. Thrust Vector Control

An analytical study of the effects of thrust vector control (TVC) by secondary gas injection on an overall engine system was performed for the AJ-1 engine. Secondary gas injection was considered because this type of system imposes more severe requirements on auxiliary components than do systems utilizing gimbals, jet tabs, etc., and has greater performance than a secondary liquid injection system. The purpose of the study was to determine whether any unusual auxiliary

VI, B, Steady-State Engine-System Analysis (cont.)

component problems are created by the size or pressure of the engine system. Reference to the latest available study of TVC by secondary gas injection [1] provides a reasonable basis for this study. This referenced report and most applicable experimental data available are based on conventional deLaval nozzles and recommend injection at the skirt exit for maximum performance. Previous Aerojet-General studies [2], however, indicated that from an overall vehicle standpoint, injection just below the end of the internal expansion sections (IES) results in better TVC performance with the forced-deflection nozzle. In order to measure the effects of the TVC systems on the engine, two methods of secondary gas injection were adopted. Schematics of these systems are shown in Figures VI-B-7 and -8.

In the first system, a small amount of hydrogen is tapped off the chamber cooling jacket to regeneratively cool the forced-deflection skirt thereby raising the hydrogen temperature to about 1000°R. This hydrogen is then available for TVC through injection directly into the main stream at the end of the IES (amplification factor 2.5 for vernier control). When not used for TVC, this hydrogen is directed to the thrust chamber injector. For heavy steering, the hydrogen is directed to a gas generator where it is combusted with a suitable amount of liquid oxygen (mixture ratio 0.24 to 6) tapped from the pump discharge (amplification factor 2.0). To emphasize the possible severity of the effect of size on TVC component design, a single gas generator and injector port is utilized for each quadrant. A summary of the most significant engine specifications for a range of TVC fuel and oxidizer flow rates is shown in Table VI-B-3. These specifications were determined by use of the Liquid Engine Design Program discussed earlier. It should be pointed out that these values are for steady-state operation, whereas the actual system will be in a transient state; therefore, values this severe will never be attained.

(1) A Theoretical and Experimental Study of Thrust Vector Control By Secondary Gas Injection, final report, Contract NAS 8-5070, United Aircraft Corporation (C).

(2) Design Study of Large Unconventional Liquid Propellant Rocket Engines and Vehicles, Aerojet-General R-LRP-257, 18 April 1962 (C).

VI, B, Steady-State Engine-System Analysis (cont.)

The second system shown is a simplified system. It uses turbine exhaust gas exclusively as the secondary gas injectant, thereby eliminating auxiliary LO_2/LH_2 gas generators, as well as any need for handling gas at 6000°R. The details of this system and the associated auxiliary component requirements are discussed in Appendix E. A summary of the effects of this system on the overall engine specifications is provided in Table VI-B-4. These effects are to be considered in the same way as for those of the first system.

b. Thrust Control

Thrust control differs from thrust modulation in that its purpose is simply to maintain the specified thrust level over the entire duration of engine firing. The magnitude of the thrust-control problem for the large high-pressure engine has been indicated in the discussion of vehicle-acceleration effects (Section VI,B,2,b) showing thrust increasing as much as 17% if no compensating control is provided.

There are four basic means of controlling thrust increase from the high net positive suction head (NPSH) resulting from acceleration of the vehicle: (1) control of the suction valves, (2) throttling of the pump inducer, (3) control of pump head-rise, and (4) modulation of valves downstream of the pump.

The use of modulating suction valves might be feasible for the bottom-tank propellant where the pump is only a short distance from the tank. The upper tank, which usually provides the more severe increase in NPSH, would require that the suction valve be located at or near the pump, rather than at the tank. This would not provide emergency shut-off capability in the event of a leak in the suction line or bellows, unless an additional valve were installed at the tank-end of the suction line. Also, because thrust modulation would be the only reason for requiring modulating capability in the suction valves, this method does not appear to be desirable.

Most of the pumping systems of the engines studied under this program utilize inducer stages driven by an independent hydraulic turbine that uses recirculated propellant from the pump discharge as the working fluid. Throttling the propellant flow to the inducer turbine would provide a means of controlling head rise through the inducer and, hence, the effective NPSH of the pump. This method is limited because the maximum possible reduction of NPSH is equal to the design head of the inducer. Although this is the most promising of the four methods, it has not been studied in detail because it entails major components rather than the auxiliary components with which this program is concerned.

The third system considered, controlling head rise through the pump, would entail the use of a shroud valve or similar device to vary the clearance between the pump-impeller housing to control pump efficiency. As with inducer throttling, this system is not concerned with auxiliary components and has, therefore, not been analyzed further.

Finally, thrust control may be achieved by the use of modulating valves in the high-pressure portion of the engine system downstream of the pump. If pump discharge valves are used, such as the ring-gate type discussed in Section VII, D, 1, these may be throttled to reduce pump discharge pressure to design values. If separate primary combustor and thrust-chamber valves are used, they may be modulated in the same way. While no detailed analysis was made, such as that performed for thrust modulation, it also appears feasible to modulate only flow to the primary combustors to lower the energy level of the turbine-inlet gas and thereby control pump output.

It may be seen that thrust control, if not achieved within the pumping system proper, will require modulating capability of either suction or discharge valves. This requirement will exist, regardless of whether or not thrust modulation or propellant utilization (Section VI, B, 3, d) systems are considered for a particular vehicle application.

VI, B, Steady-State Engine-System Analysis (cont.)

c. Thrust Modulation

A study of thrust modulation was performed on the AJ-1 engine to determine a means for throttling a large high-pressure engine over a thrust range of 10:1. Three methods of modulation were analyzed by utilizing the IBM 7094 liquid engine design computer program previously discussed. For purposes of this study, the resistance of a component is defined by the following function:

$$C = \rho \frac{\Delta P}{\dot{w}^2}$$

where

C = resistance, sec^2/ft^5

ρ = fluid density, lb/ft^3

ΔP = pressure drop, lb/ft^2

$\dot{w}$ = fluid flow rate, lb/sec

and because

$$\dot{w} = \rho AV$$

where

A = cross-sectional flow area, ft^2

V = fluid velocity, ft/sec

C is related to A for a fixed ΔP, ρ, and V

$$C \sim \frac{1}{A^2}$$

In the first system, the resistance in the oxidizer primary combustor line is increased, thus reducing the primary combustor mixture ratio.

VI, B, Steady-State Engine-System Analysis (cont.)

This results in lowering of the thrust by generating a less energetic turbine drive gas. Thrust-chamber mixture ratio is allowed to vary for the simplicity of the system. In this system, both the valve and injector were considered as variable area components. For a very wide range of throttling, the injector area must be variable so that sufficient velocity exists through the injector to maintain the desired combustion efficiency. The means of acquiring variable injector area was not studied as the injector is considered to be a major, rather than an auxiliary, component. The area-variation requirements for throttling can be determined from Figure VI-B-9 which shows the value of total resistance required from the valve and injector as a function of thrust. A 4:1 throttling range, which is the practical limit of this throttling method, requires a resistance change of 0.63 to 5.7 or a range of 9.05:1. This corresponds to a total-area change of 3:1, which is easily attained in a single valve. If the injector were not a variable-area component, the pressure drop across the injector at 1.5×10^6-lb thrust would decrease to 20 psia from a design value of 255 psia. This method of thrust modulation is not satisfactory because it is capable of achieving only 1/4 thrust while requiring an additional component (the oxidizer primary combustor valve). In addition, the thrust-chamber mixture ratio has increased (at 1.5×10^6-lb thrust) to 7.24, which significantly lowers the engine specific impulse and increases the combustion temperature in the main chamber. For this system, Figure VI-B-10 shows the pressure distribution throughout the engine as a function of thrust. The figure clearly shows that at low thrust levels there is insufficient pressure drop available between the pump discharges and the primary combustor for efficient primary combustor injector operation.

The second system evaluated consisted of throttling the two main propellant discharge valves so a constant engine mixture ratio is maintained. This eliminates the temperature problem in the main chamber and the need for an extra component (oxidizer primary combustor valve), but results in limited throttling

VI, B, Steady-State Engine-System Analysis (cont.)

capability. Figure VI-B-11 shows valve resistance as a function of thrust level and shows a throttling capability of less than 3:1. For throttling of 2:1, the maximum resistance range is 143:1 (for the fuel valve). This represents an area change of 12:1 for only 2:1 throttling. This system was not investigated further.

The final system studied consisted essentially of a combination of the first two systems. This system throttles the main discharge valve and the primary injector on the fuel side, the primary injector in the oxidizer primary combustor circuit, and the thrust-chamber valve and injector in the oxidizer-liquid circuit. Both primary combustor and thrust chamber mixture ratios are maintained constant at their design values. Figure VI-B-12 shows the results of this study in terms of the required resistance in each of the three circuits described above. This method has the capability of throttling over a range of more than 25:1 while maintaining sufficient pressure drop (See Figure VI-B-13) throughout the engine for efficient operation. The ranges of resistance represented are 277:1, 408:1, and 462:1 for the fuel circuit, oxidizer-liquid circuit, and oxidizer primary combustor circuit, respectively. The maximum area change (oxidizer primary combustor circuit) is 21.5:1. Current work on variable-area valves and injectors that adequately covers the range is being conducted in Contract AF 04(611)-9100 by Space Technology Laboratories for Edwards Air Force Base and in company-funded work at Aerojet-General. Figure VI-B-13 shows the pressure distribution throughout the engine as a function of thrust for the throttling system. From a pressure-availability standpoint, as compared with the previous systems this system is the best, and more than adequate for efficient injector operation. The primary drawback of this system is the controls requirement for six variable-area components. Table VI-B-5 summarizes the major operating conditions at full thrust and approximately 1/10 full thrust.

The study has shown that there is a definite need on large high-pressure rocket engines for several types of modulating valves with considerable area-variation capability. The special problems created in these valves because of their size and pressure are discussed at length in Section VII of this report.

VI, B, Steady-State Engine-System Analysis (cont.)

d. Propellant Utilization (PU)

A propellant-utilization control system regulates engine flow rates to achieve simultaneous emptying of the propellant tanks. It consists of three basic components: (1) a sensing device to monitor flow rates from the tanks (such as a flow meter and/or liquid level device); (2) a control system to detect and interpret flow-rate variations; and (3) a PU valve that on receiving an appropriate signal from the control system, will adjust the engine mixture ratio.

The purpose of this investigation was to determine design problems associated with a propellant-utilization control valve in a high thrust, high chamber-pressure engine. Because the location of the valve within the system greatly influences the valve-design requirements, the determination of the best locations was an important part of the program.

The AJ-1 engine was selected for study and is shown schematically in Figure VI-B-14. The sketch shows the possible valve locations that were considered. Arbitrary resistances at these locations were incorporated into the steady-state computer program to vary the design-point mixture ratio $\pm 10\%$, i.e., from 5.4 to 6.6. With the exception of the oxidizer secondary-combustor line (Location 7), control over the entire range required two valves. For lower mixture ratios, control was achieved with a valve in the oxidizer line. For the higher mixture ratios, a control valve was required somewhere in the fuel line. Both valves are full open at the design mixture ratio. Partial closure of either valve resulted not only in a change of the engine mixture ratio, but corresponding changes in thrust, chamber pressure, pump discharge pressure, and primary combustor temperature as well. The magnitude of these parameter changes served as criteria to evaluate the effectiveness of the location. The results are shown in Figures VI-B-15 to -19.

The results show that for this situation the most effective locations are the second oxidizer and fuel-pump by-passes (Locations 5 and 6, respectively). The flow rates associated with these locations are shown in Figures VI B-20 and -21. Table VI-B-6 is a tabulation of the more vital information necessary for the by-pass valve designs. It includes the fluid properties, flow rates, pressure drops, K-factor values, permitted leakage, and flow-rate tolerances. The latter is based on a maximum allowable mixture-ratio variation of ± 0.025. This means that if the mixture ratio varies ± 0.025, the by-pass flow rates will change no more than 5% for the LO_2 bypass valve and 4% for the fuel bypass valve. The amount of leakage permitted shall be no greater than 5% and 4% of the maximum LO_2 and LH_2 by-pass flow rates, respectively.

Regardless of the wide variations that result when the valve is placed in the oxidizer-secondary combustor line (Location 7), this location is potentially the best. First of all, PU control would require only one valve in a line with plenty of available resistance. Secondly, there would be no need of pump by-pass lines. By-pass lines necessitate higher pump flow rates and, hence, greater pump weight; these disadvantages would also be eliminated.

However, to make Location 7 effective, its influence on the previously-cited engine parameters must be reduced. Actually the wide variations in thrust and chamber pressure that occur when the valve is in Location 7 result from the effect of the latter on the primary combustor mixture ratio. Possibly another valve, located in the oxidizer line just upstream of the primary combustor, could be operated in conjunction with the valve at Location 7. The additional valve would serve to keep the primary combustor mixture ratio constant while the other valve is varying the engine mixture ratio. This possibility would require a supplementary investigation.

Valves in the suction lines (upstream of the pumps) were not considered in great detail because the literature was fairly unanimous in citing these locations as undesirable. A few computer runs were made with valves assumed at these locations. The results shown in Figure VI-B-14 verify the ineffectiveness of these placements.

VI, B, Steady-State Engine System Analysis (cont.)

Valve locations in the hot-gas stream would have offered other interesting possibilities had the turbines been independent of one another. An excellent unpublished Aerojet-General report* on propellant utilization in the M-1 engine showed that turbine bypass valves proved to be very effective in regulating the mixture ratio while keeping variations in the engine operating point to a minimum. However, because the AJ-1 turbines are mechanically linked, the pump speeds cannot be independently controlled to change the mixture ratio. Therefore, hot-gas locations were not considered in this analysis.

The PU investigation is summarized below:

1. For a two-valve PU system, mixture-ratio control can be most effectively achieved with pump by-pass valves with the bypass circuits as far downstream from the pumps as possible.

2. A single control valve in the oxidizer-secondary combustor line is potentially the best, pending additional research.

3. The nature of the problem is such that the PU valve will have design requirements and problems different than the suction and discharge valves in the AJ-1. For instance, the valve may never be completely closed, if so, the sealing requirements may be different. In addition, the valve will have to be of a type suitable for flow regulation.

e. Tank Pressurization

A tank-pressurization system was incorporated into the AJ-1 steady-state computer program to provide design criteria for a heat exchanger.

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* Steady State Analysis of AJ-1000A Rocket Engine for Proposal LR61066, Aerojet-General Memo D. E. Glum to D. E. Price, 27 September 1961 (Conf.)

VI, B, Steady-State Engine System Analysis (cont.)

In this section, tank pressurization is briefly discussed from an overall systems point of view. Of chief concern is the determination of the best heat-exchanger location and the conditions at which the heat exchanger must operate. A thorough discussion of the actual heat-exchanger-design criteria is included in the Heat Exchanger Section (IX).

There are a number of ways to pressurize propellant tanks. The more common ways are (1) to use small quantities of propellants heated to a gas; (2) to use an auxiliary inert-gas system, and (3) to use hot turbine-exhaust gases. The system selected for this study utilized the first approach; although it is not necessarily optimum, it does provide rigid design requirements for heat exchangers.

The chosen flow rates, based upon M-1 engine design criteria, were 1 and 2% of the total oxygen and hydrogen flow rates, respectively. Propellants are tapped off the pump-discharge lines, heated, and ducted to the propellant tanks at a temperature of 960°R for oxygen and 560°R for hydrogen.

The heat exchanger was designed to heat both the LO_2 and LH_2 at the same time with primary combustor exhaust gases serving as the heat source. The effect of the heat exchanger on the design specifications of the engine was evaluated at two locations; the turbine inlet and the turbine exit.

The calculated engine design points for each location are shown in Table VI-B-7, where the basic AJ-1 engine is included for comparison. The values of 20, 45, and 100 psi refer to assumed pressure drops across the heat exchanger.

In general, the results show that either location is acceptable and neither severely effects the basic engine design point. For example, the increase in the required LO_2 pump discharge pressure was 241 psi (5.9%) with the heat exchanger (having a ΔP of 100 psi) located aft of the turbine and 215 psi (5.3%) when located

VI, B, Steady-State Engine System Analysis (cont.)

at the turbine inlet. However, based on experience with tank-pressurization systems, locating the heating exchanger at the turbine exit would probably prove to be less complicated. Figure VI-B-22 shows how the heat exchanger discussed in Section IX, could be installed in the AJ-1 turbine exhaust.

It was considered unnecessary to evaluate heat exchangers in a typical gas-generator cycle because requirements of the staged-combustion cycle in the AJ-1 are much more rigid.

TABLE VI-B-1

COMPONENT AND BALANCE TOLERANCES
FOR
STATISTICAL OFF-DESIGN STUDY

Parameter	Tolerance
Pump Head*	± 5%
Turbine efficiency**	± 7%
Turbine area***	± 5%
Line resistances+	± 15%
Thrust chamber area	± 1.5%
Thrust chamber I_{sp}	± 1.3%
Thrust-chamber skirt resistance	± 15%
Engine mixture ratio	± 3.5%
Primary combustor mixture ratio	± 5%

---

* At a given flow and speed
** At a given U/C ratio
*** At the nozzle throat
\+ Constant for a given line and equal to $P/\dot{w}^2$

TABLE VI-B-2

VARIATION OF ENGINE OPERATING CONDITIONS WITH LONGITUDINAL "G" LOADING

Operating Condition	1G	1.25G	6G	10G
Engine:				
Thrust (F) lb	6.0×10^6	6.1551×10^6	6.665×10^6	7.0294×10^6
Specific impulse (I_{sp}) sec	383.0	382.06	378.72	375.64
Mixture ratio (MR_E)	6.0	6.0847	6.387	6.6598
Chamber pressure (P_c) psia	2580	2650	2885	3058.83
Oxidizer flow rate ($\dot{w}_{oc}$) lb/sec	13,428	13,836	15,216	16,270
Fuel flow rate ($\dot{w}_{fc}$) lb/sec	2238	2273.94	2382	2443
Oxidizer tank pressure (P_{ot}) psia	84.6	107	382	622.9
Fuel tank pressure (P_{ft}) psia	32.0	32.9	34	28.45
Oxidizer Pump:				
Speed (N_o) rpm	6758	6881.75	7217	7463.7
Efficiency (η_o)	0.71528	0.71004	0.68591	0.66591
Suction head above vapor (H_{osv}) ft	552.6	602.86	1151.8	1626.54
Suction pressure (P_{os}) psia	289.4	314	582.78	815.18
Discharge pressure (P_{od}) psia	4115.4	4255.91	4796	5219.79
Fuel Pump:				
Speed (N_f) rpm	6758	6881.75	7216.8	7463.71
Efficiency (η_f)	0.71951	0.71997	0.72018	0.7174
Suction head above vapor (H_{fsv}) ft	513.45	542.91	601.82	397.27
Suction pressure (P_{fs}) psia	32.0	32.9	34.7	28.45
Discharge pressure (P_{fo}) psia	4521	4691.55	5159.8	5493.19
Turbine:				
Total inlet temperature(T_{tit}) °F	1440	1482.13	1625.86	1746.29
Total inlet pressure (P_{tit}) psia	3493	3604	3965.8	4235.15
Pressure ratio, total inlet to static exhaust (P_{tit}/P_{te})	1.2775	1.28202	1.29218	1.29896
Efficiency (η_T)	0.82173	0.82393	0.82968	0.83367

TABLE VI-B-2 (cont.)

Operating Condition	1G	1.25G	6G	10G
Gas Generator:				
Chamber pressure (P_g) psia	3618.6	3733.52	4108	4387
Mixture ratio (MR_G)	0.915	0.9411	1.03069	1.10576
Oxidizer Inducer:				
Speed (N_{io}) rpm	1643	1666	1710.63	1737.87
Pressure rise (ΔP_{io}) psi	204.8	207	200.78	192.28
Suction head above vapor (H_{osvi}) ft	134.284	180.04	741.74	1233.79
Turbine flow rate ($\dot{w}_{tio}$) lb/sec	2142.66	2174.86	2248.47	2298.96

TABLE VI-B-3

VARIATION OF ENGINE SPECIFICATIONS
DUE TO AUXILIARY-GAS-GENERATOR THRUST-VECTOR-CONTROL SYSTEM

$\dot{w}_o$ lb/sec	$\dot{w}_f$ lb/sec	F lb x 10^{-6}	I_{sp} sec	MR_{sc}
0	0	6.000	383.0	6.000
0	44	5.863	381.8	6.059
0	100	5.682	380.2	6.141
0	500	3.921	359.6	7.192
120	500	3.932	360.3	7.148
1100	500	4.027	366.1	6.789
2256	500	4.136	373.0	6.373
2500	500	4.366	379.6	5.993
2750	500	4.480	384.2	5.717
3000	500	4.531	386.4	5.588

$\dot{w}_o$ = Oxidizer flow rate

$\dot{w}_f$ = Fuel flow rate

F = Resultant engine thrust

I_{sp} = Resultant specific impulse

MR_{sc} = Secondary combustor (thrust chamber) mixture ratio

NOTE: Primary combustor mixture ratio (MR_{pc}) is maintained constant by throttling of the oxidizer primary-combustor line.

TABLE VI-B-4

VARIATION OF ENGINE SPECIFICATIONS
DUE TO TURBINE-EXHAUST THRUST-VECTOR-CONTROL SYSTEM

$\dot{w}$ lb/sec	F lb x 10^6	I_{sp} sec	MR_{sc}	MR_{pc}
0	6.000	383.0	6.000	0.915
50	5.897	381.9	6.057	0.916
100	5.793	380.7	6.118	0.918
500	4.839	369.0	6.739	0.931
620	4.488	364.0	7.001	0.935

$\dot{w}$ = TVC turbine exhaust gas flow rate

F = Resultant engine thrust

I_{sp} = Resultant specific impulse

MR_{sc} = Secondary combustor (thrust chamber) mixture ratio

MR_{pc} = Primary combustor (gas generator) mixture ratio

TABLE VI-B-5

ENGINE AJ-1 - PERFORMANCE AT 1/10 THRUST

Engine Parameter	Thrust	
	6×10^6 lb	0.568×10^6 lb
Specific impulse, sec	383	371
Engine mixture ratio	6.0	6.0
Thrust chamber pressure, psia	2500	242
Oxidizer flow rate, lb/sec	13428	1313
Fuel flow rate, lb/sec	2238	219
Turbopump speed, rpm	6758	3963
Oxidizer-pump efficiency, %	71.5	50.7
Oxidizer-pump discharge pressure, psia	4071	1600
Fuel-pump efficiency, %	71.9	34.1
Fuel-pump discharge pressure, psia	4521	1492
Turbine-pressure ratio	1.28	1.28
Turbine efficiency, %	82.2	60.8
Turbine-inlet temperature, °F	1440	1440
Primary-combustor mixture ratio	0.915	0.915
Oxidizer-inducer speed, rpm	1643	977
Oxidizer-inducer head, ft	417	237
Oxidizer inducer efficiency, %	58.7	14.3
Inducer-turbine flow rate, lb/sec	2143	1316
Inducer-turbine efficiency, %	57.0	56.2

TABLE VI-B-6

PROPELLANT UTILIZATION
DESIGN REQUIREMENTS FOR PU VALVES
LOCATED IN BY-PASS LINES

	FLUID PROPERTIES			DESIGN REQUIREMENTS			
Valve	Density lb/ft^3	Temp °F	Max Flow Rate lb/sec	ΔP @ Max Flow Rate lb/in^2	K-Factor @ Max Fl-Rate $lb/sec\sqrt{psi}$	Flow Rate Tolerance %	Permitted Leakage % Max Fl-Rate
Oxidizer							
Bypass ⑤ (LO_2)	70.5	-290	1530	2660	30	± 5.0*	5.0
Fuel							
Bypass ⑥ (LH_2)	4.4	-420	253	3718	4.2	± 4.0*	4.0

*Based on maximum, allowable mixture-ratio variation of ± 0.025.

TABLE VI-B-7

EFFECT OF TANK PRESSURIZATION SYSTEM ON AJ-1 ENGINE PARAMETERS

System Description	Heat Exchanger at Turbine Inlet			Heat Exchanger at Turbine Exit			AJ-1 Basic Engine Parameter Values
Pressure drop across heat exchanger, psi	20	45	100	20	45	100	
ENGINE							
Thrust, lb	6×10^6	6×10^6	6×10^6	6×10^6	6×10^6	6×10^6	6×10^6
Specific Impulse, sec	383	383	383	383	383	383	383
Mixture Ratio	6.0	6.0	6.0	6.0	6.0	6.0	6.0
Chamber Pressure, psia	2500	2500	2500	2500	2500	2500	2500
Oxidizer Flow Rate, lb/sec	13585	13586	13588	13585	13586	13588	13428
Fuel Flow Rate, lb/sec	2283	2283	2283	2283	2283	2283	2238
Heat Exch. Oxid. Flow Rate, lb/sec	157.8	158.4	159.8	157.7	158.5	160.2	-
Heat Exch. Fuel Flow Rate, lb/sec	41.5	45.2	45.6	45.1	45.3	45.7	-
PUMP							
Discharge Pressure, LO_2, psia	4179	4212	4286	4179	4220	4312	4071
Discharge Pressure, LH_2, psia	4582	4614	4689	4581	4623	4715	4521
Specific Speed, LOX, $\frac{rpm(gpm)^{\frac{1}{2}}}{(ft)^{3/4}}$	2566	2560	2546	2566	2558	2541	2560
Specific Speed, LH_2, $\frac{rpm(gpm)^{\frac{1}{2}}}{(ft)^{3/4}}$	434	433	431	434	433	430	430
Shaft Speed, rpm	6817	6838	6885	6817	6844	6902	6758
Flow Rate, LO_2, lb/sec	15745	15754	15774	15745	15757	15782	15570
Flow Rate, LH_2, lb/sec	2283	2283	2283	2283	2283	2283	2238
TURBINE							
Mixture Ratio	0.915	0.915	0.915	0.915	0.915	0.915	0.915
Flow Rate, lb/sec	4286	4286	4286	4286	4286	4286	4286
Turbine Efficiency, %	81.9	81.9	81.9	81.9	81.9	81.9	82.2
Turbine Pressure Ratio	1.293	1.295	1.300	1.290	1.292	1.299	1.2775
Turbine Inlet Pressure, psia	3531	3538	3551	3551	3591	3680	3493
Turbine Exit Pressure, psia	2732	2732	2732	2754	2779	2834	2734
Turbine Inlet Temp, °F	1423	1423	1423	1440	1440	1440	1440
PRIMARY COMBUSTOR							
Pressure, psia	3679	3712	3786	3679	3720	3812	3619
Temperature, °F	1440	1440	1440	1440	1440	1440	1440

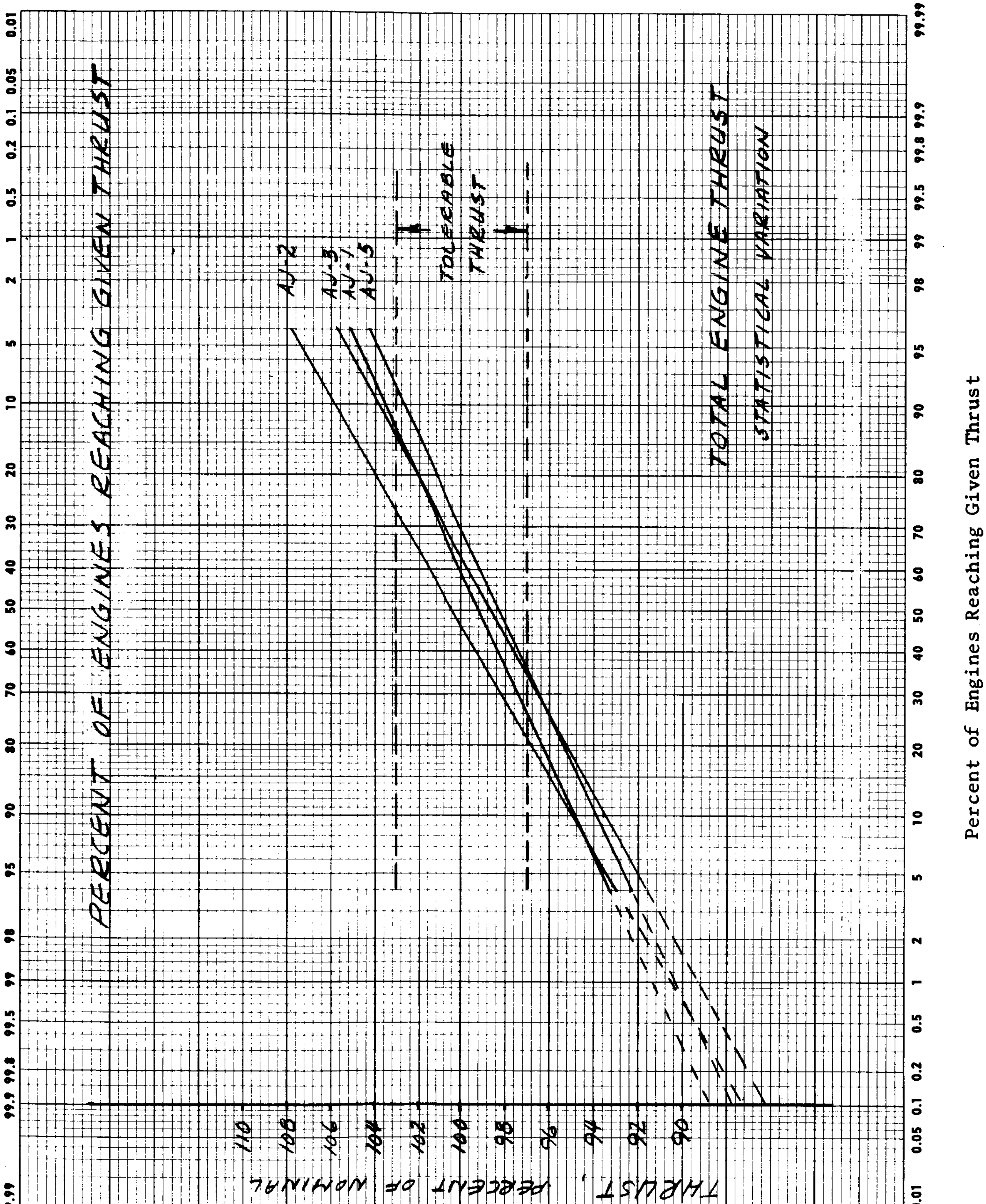

PERCENT OF ENGINES REACHING GIVEN THRUST
TOTAL ENGINE THRUST
STATISTICAL VARIATION
TOLERABLE THRUST
AJ-2
AJ-3
AJ-5
110 108 106 104 102 100 98 96 94 92 90
THRUST, PERCENT OF NOMINAL
Percent of Engines Reaching Given Thrust

Figure VI-B-1

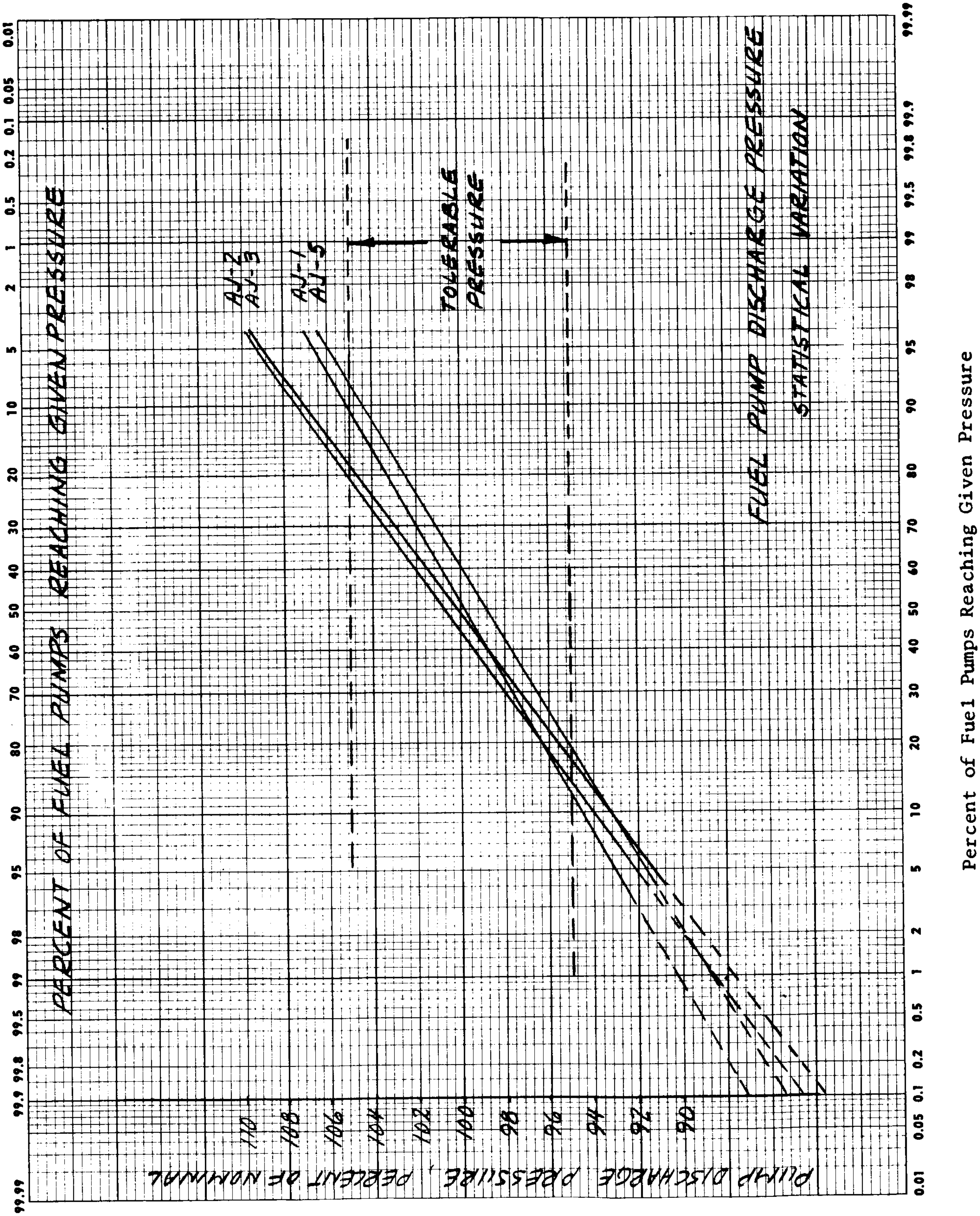

PERCENT OF FUEL PUMPS REACHING GIVEN PRESSURE
AJ-2
AJ-3
AJ-1
AJ-5
TOLERABLE PRESSURE
FUEL PUMP DISCHARGE PRESSURE
STATISTICAL VARIATION
PUMP DISCHARGE PRESSURE, PERCENT OF NOMINAL
110
108
106
104
102
100
98
96
94
92
90
Percent of Fuel Pumps Reaching Given Pressure

Figure VI-B-2

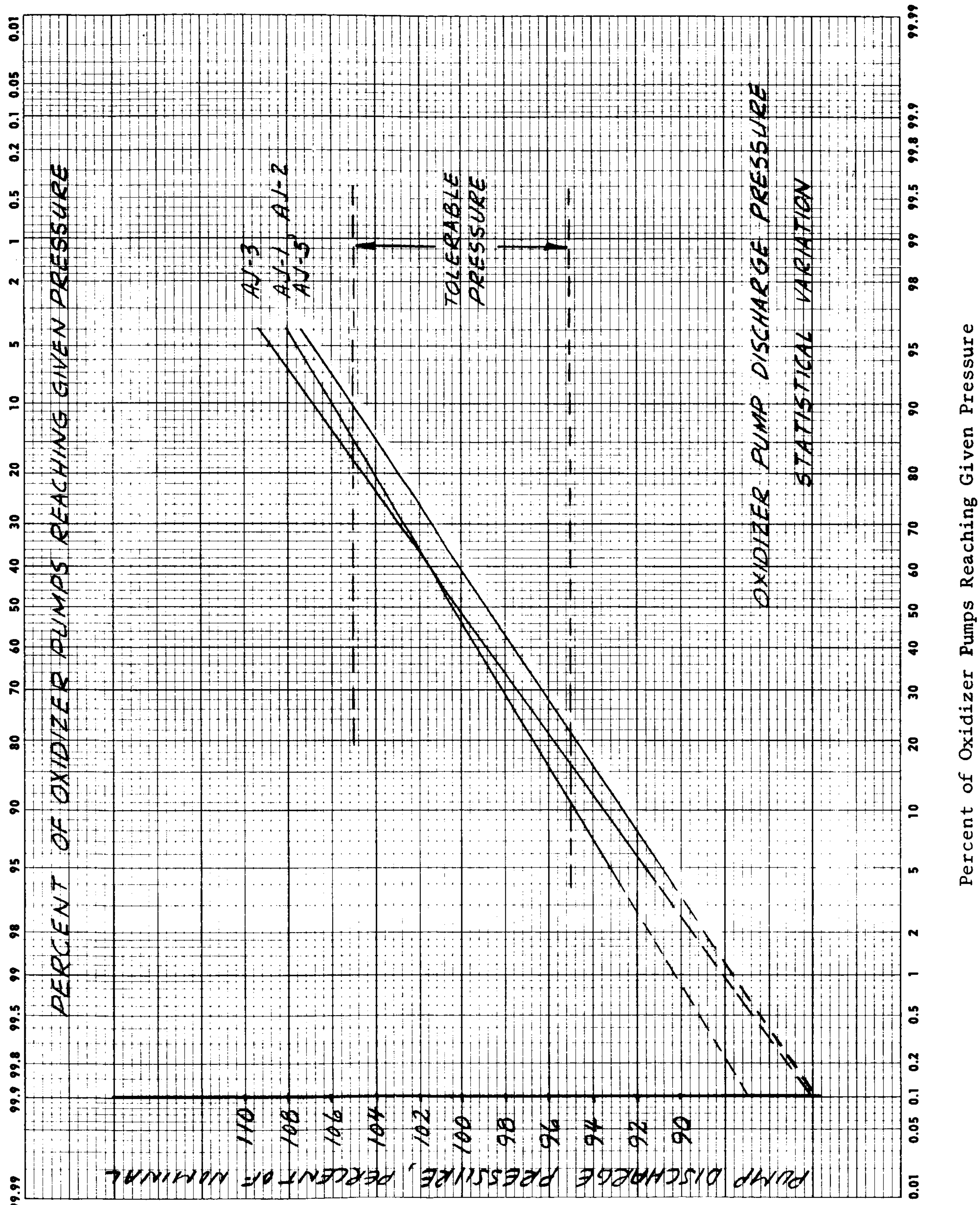
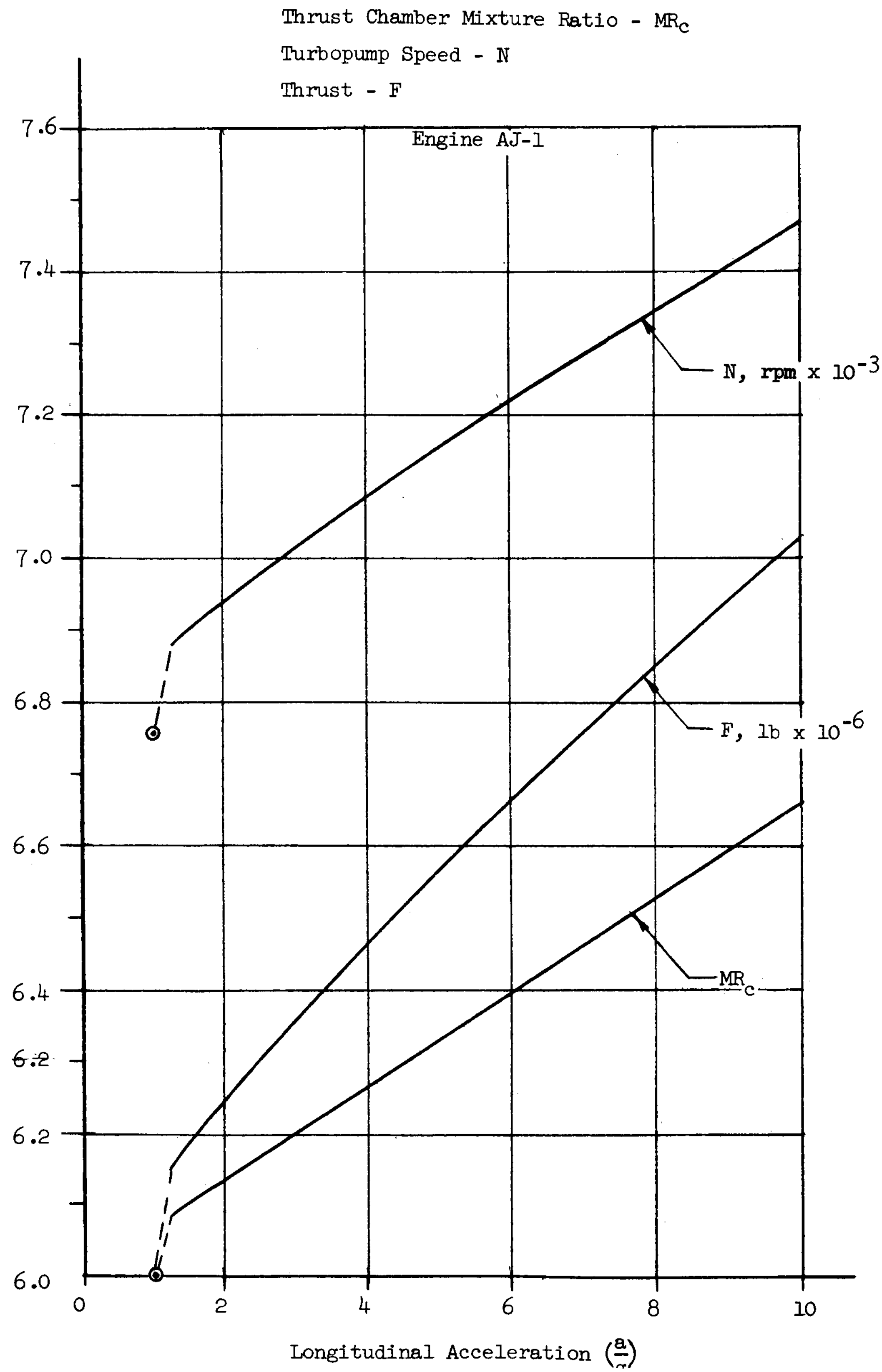

PERCENT OF OXIDIZER PUMPS REACHING GIVEN PRESSURE
AJ-3
AJ-1 AJ-2
AJ-5
TOLERABLE PRESSURE
OXIDIZER PUMP DISCHARGE PRESSURE
STATISTICAL VARIATION
PUMP DISCHARGE PRESSURE, PERCENT OF NOMINAL
110
108
106
104
102
100
98
96
94
92
90
Percent of Oxidizer Pumps Reaching Given Pressure

Figure VI-B-3

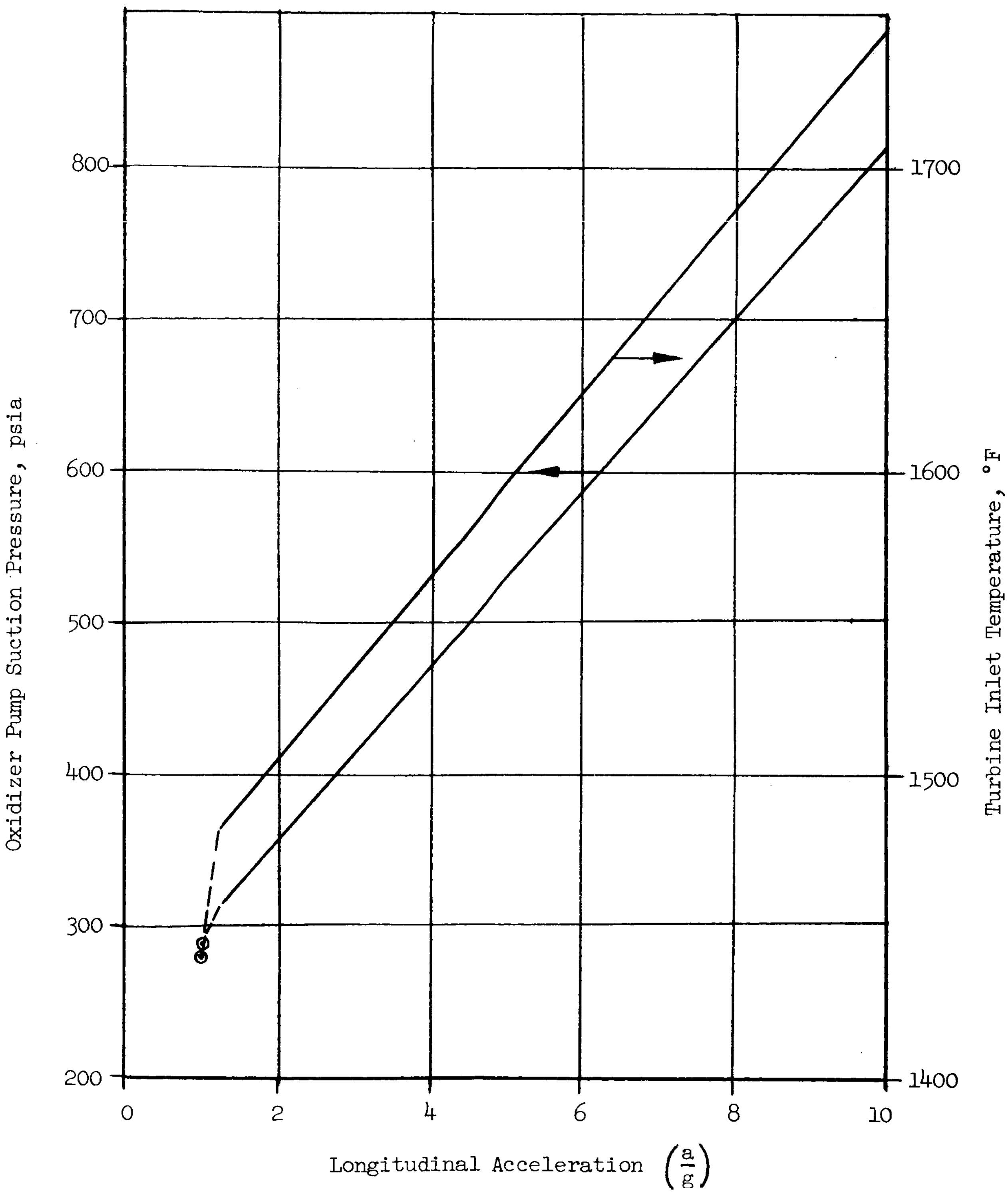


Variations of Engine Specifications vs G Loading

Figure VI-B-4

Report 5329-F

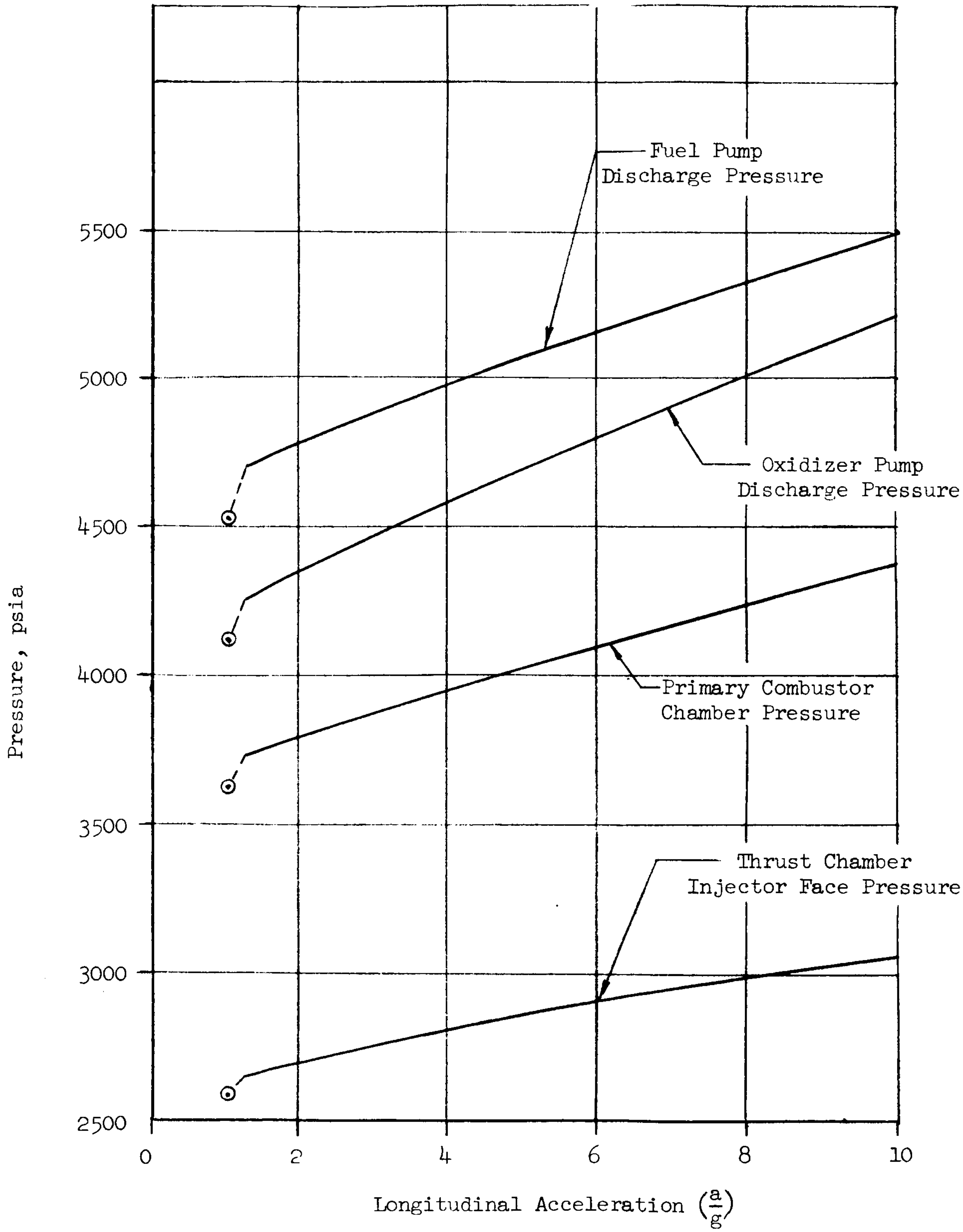
Engine AJ-1



Variations of Engine Specifications vs G Loading

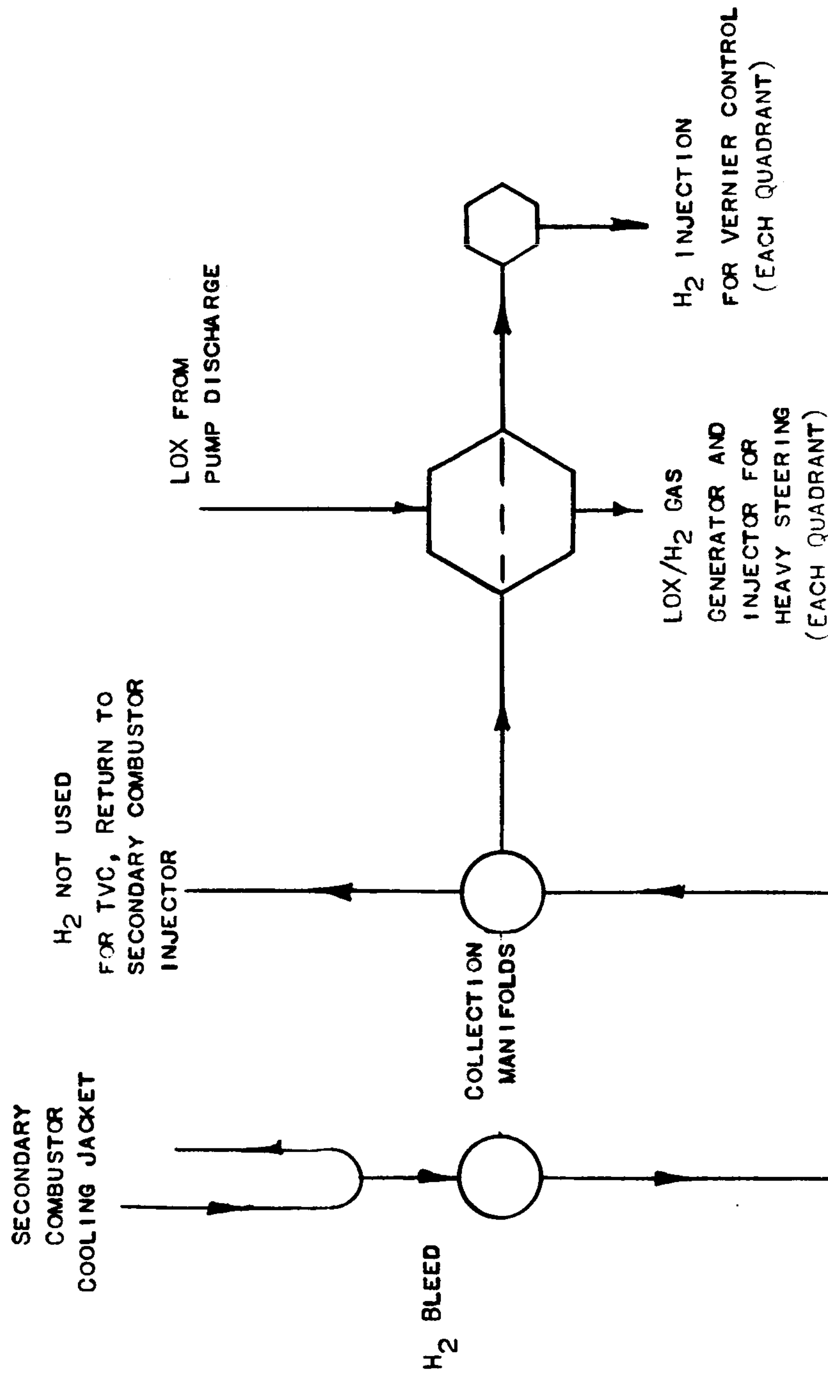
Figure VI-B-5

Engine AJ-1



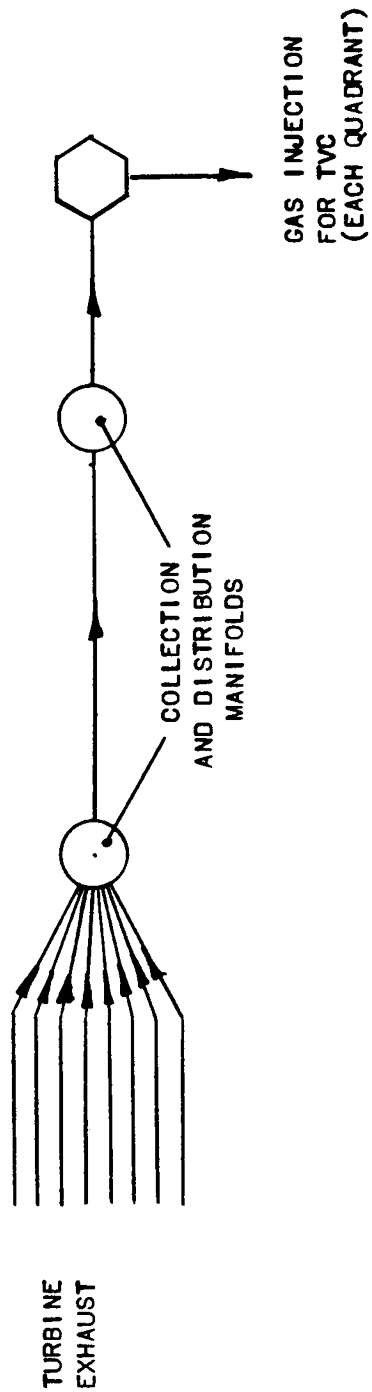
Variations of Engine Specifications vs G Loading

Figure VI-B-6



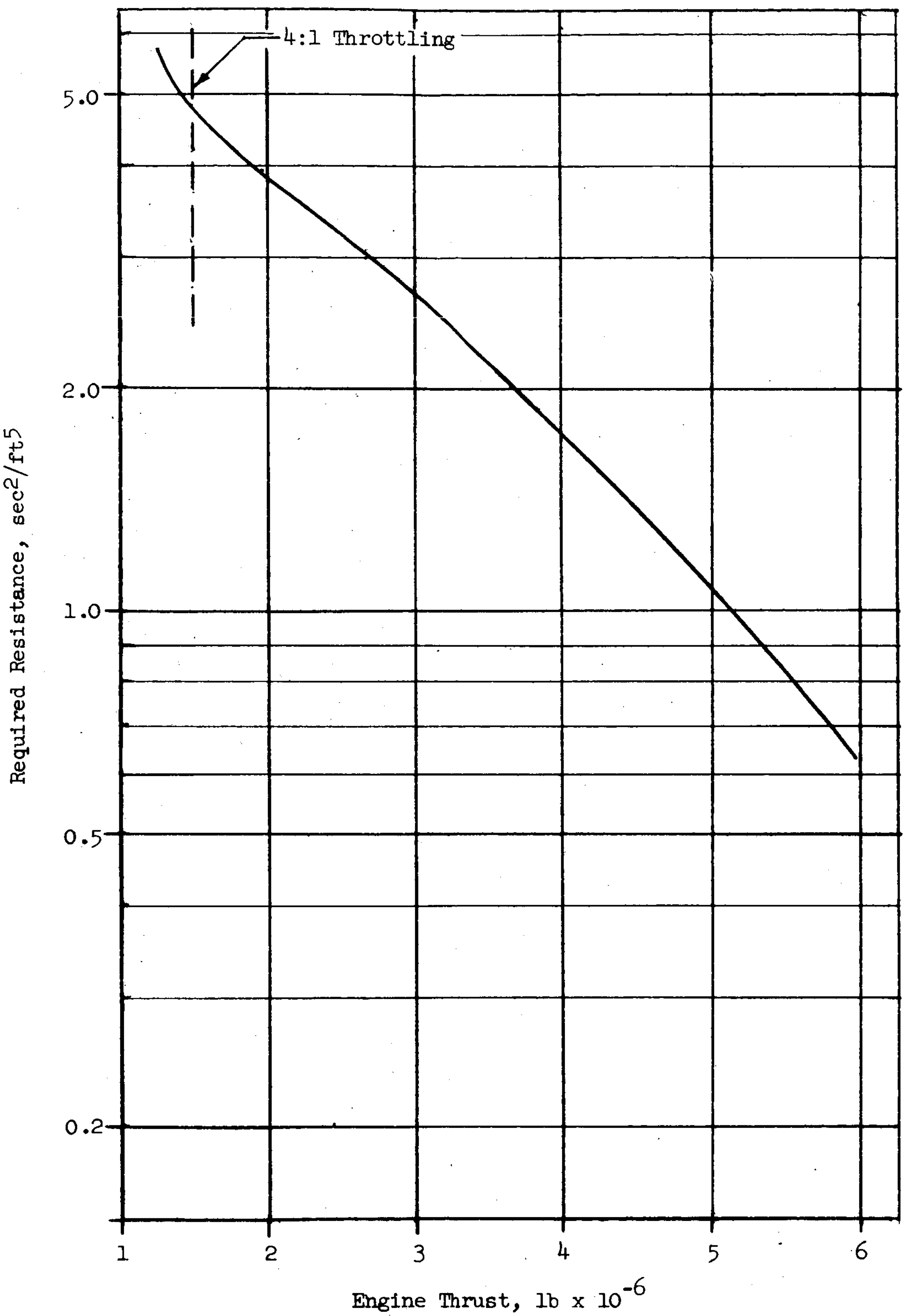
Thrust Vector Control System Flow Schematic (Gas Generator)

Figure VI-B-7



Thrust Vector Control System Flow Schematic (Turbine Exhaust)

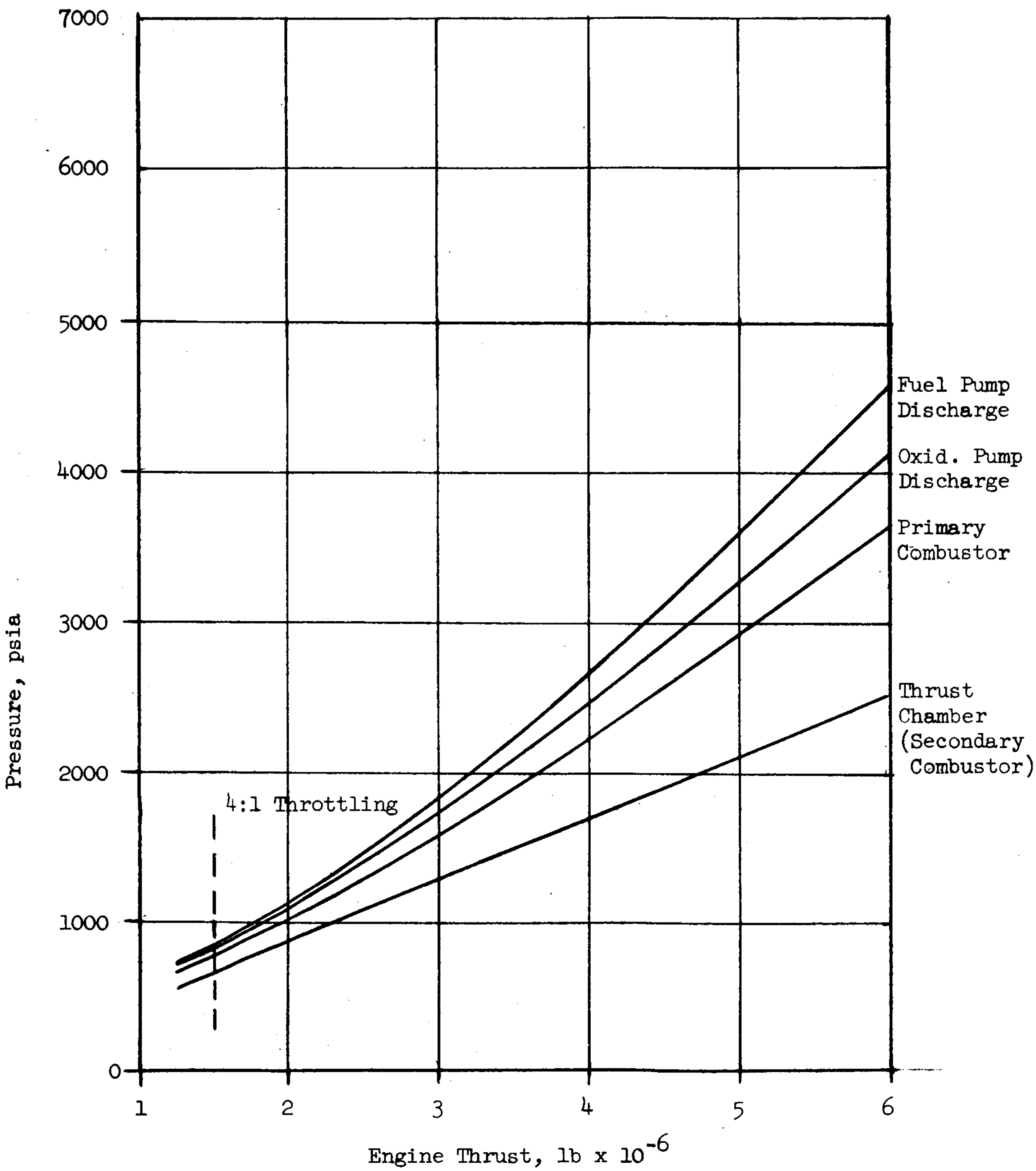
Figure VI-B-8



Total Resistance Required, Engine Throttling by Oxidizer Primary Combustor Valve and Injector

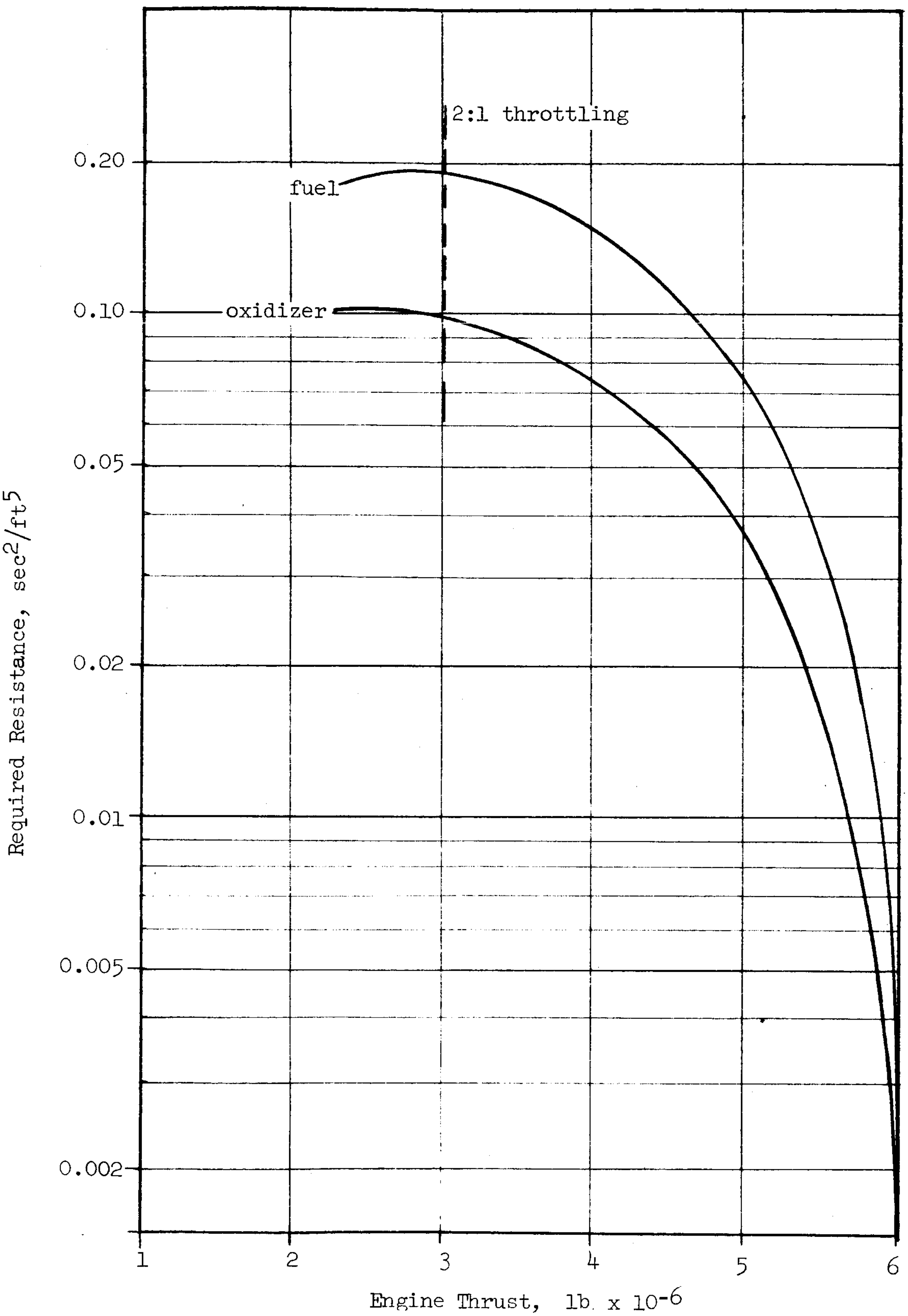
Figure VI-B-9

Engine Throttling by
Oxidizer Primary Combustor
Valve and Injector



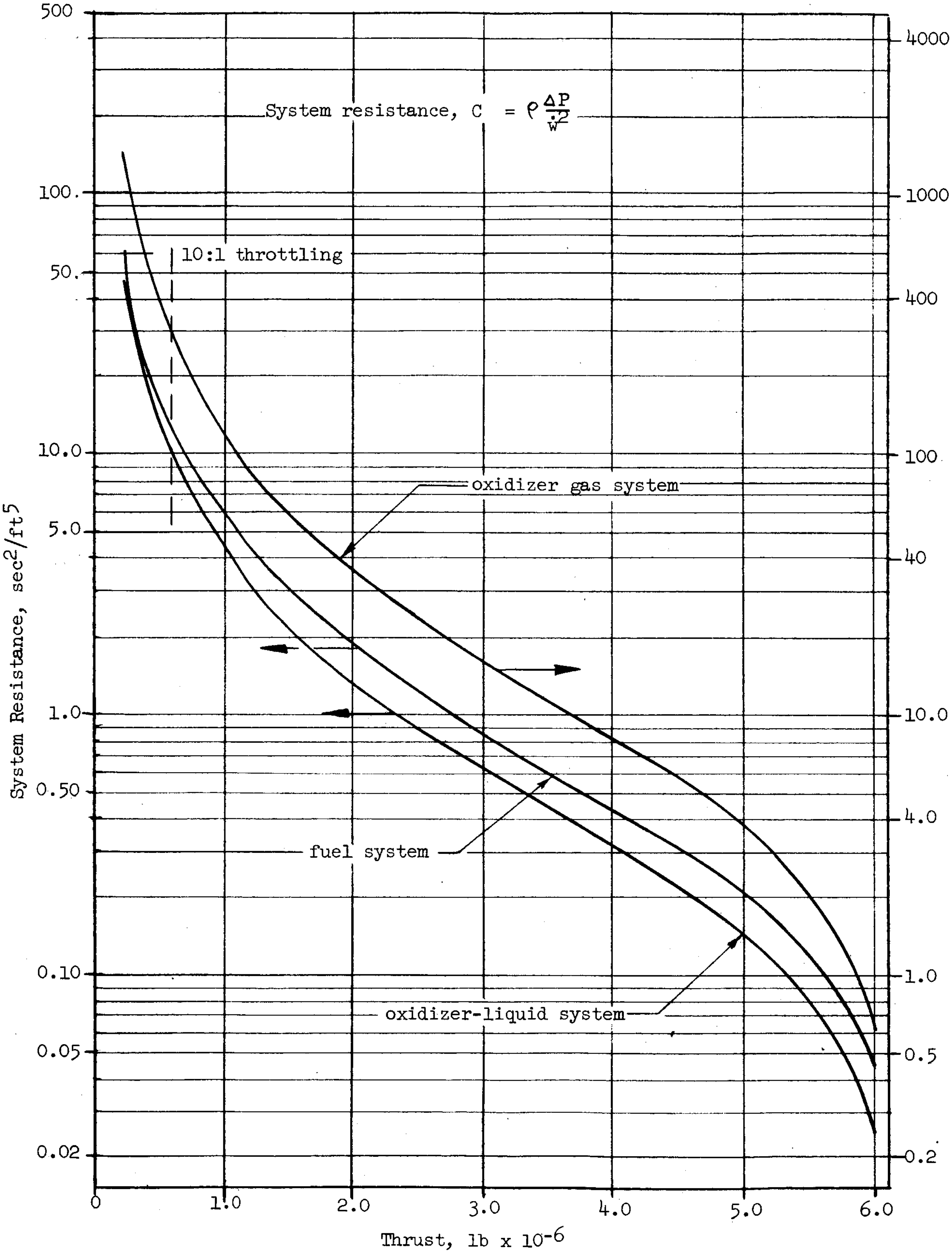
Engine Pressure Distribution

Figure VI-B-10



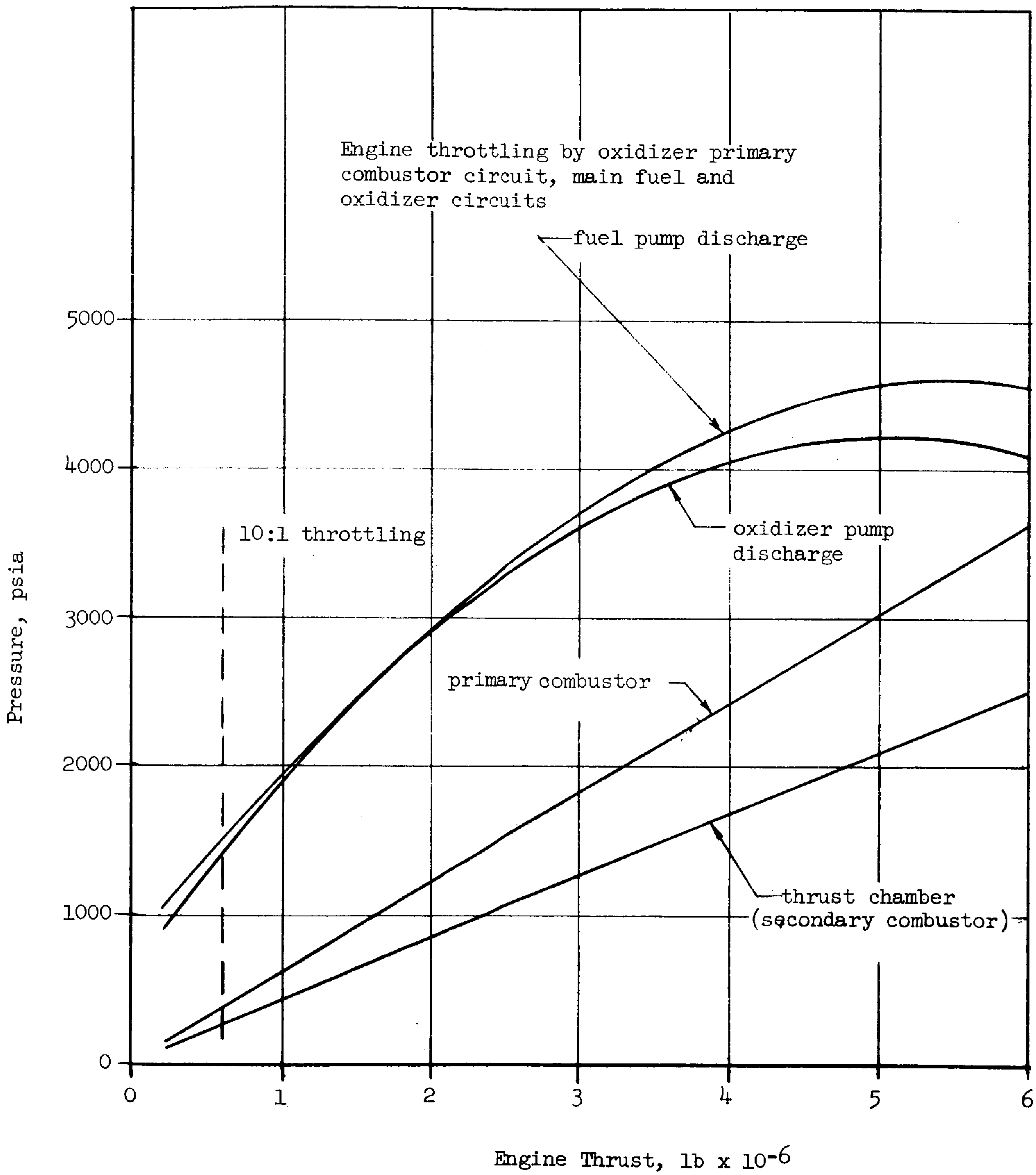
Resistance Required, Engine Throttling by Main Propellant Discharge Valves

Figure VI-B-11



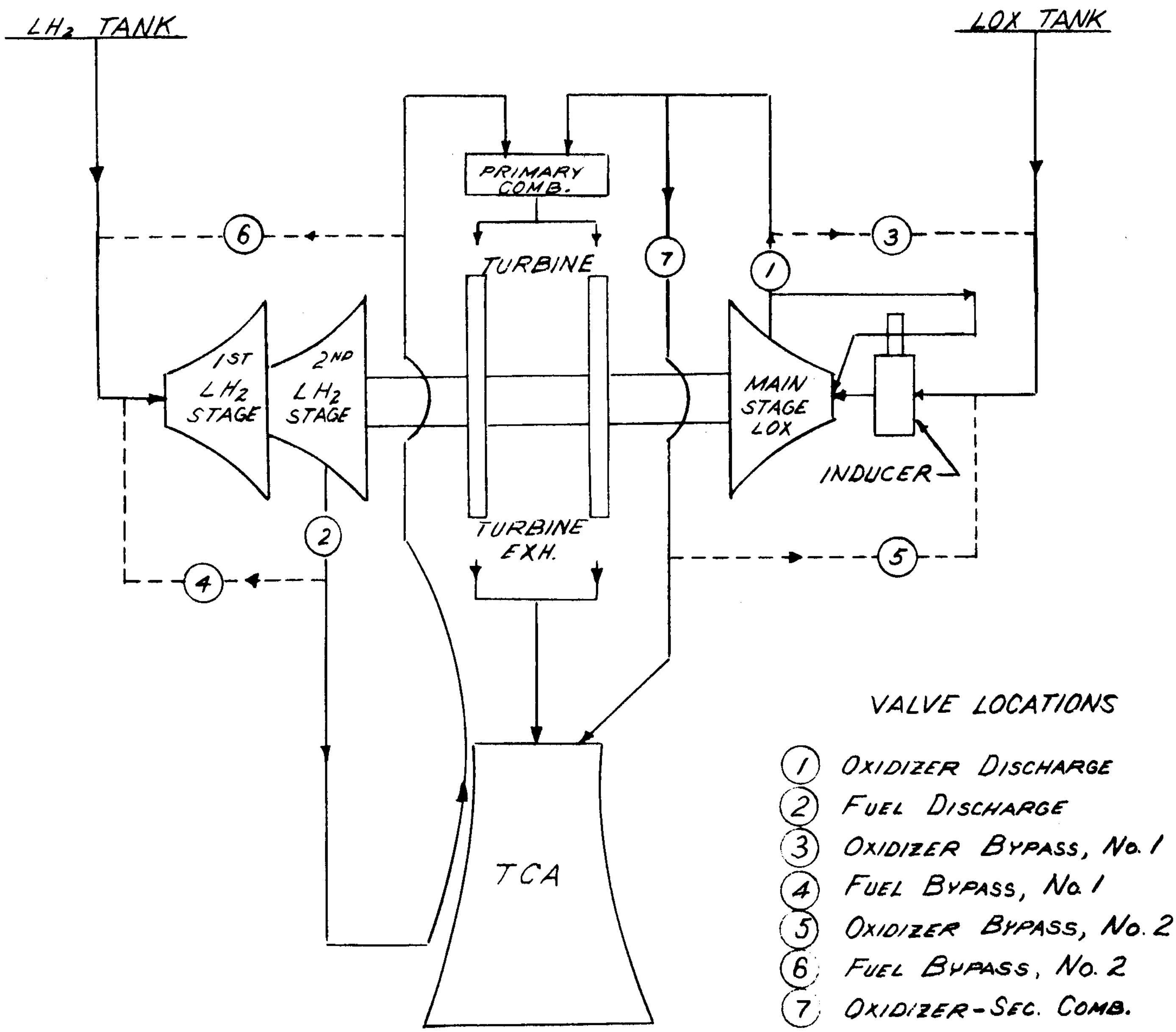
System Resistance vs Thrust

Figure VI-B-12



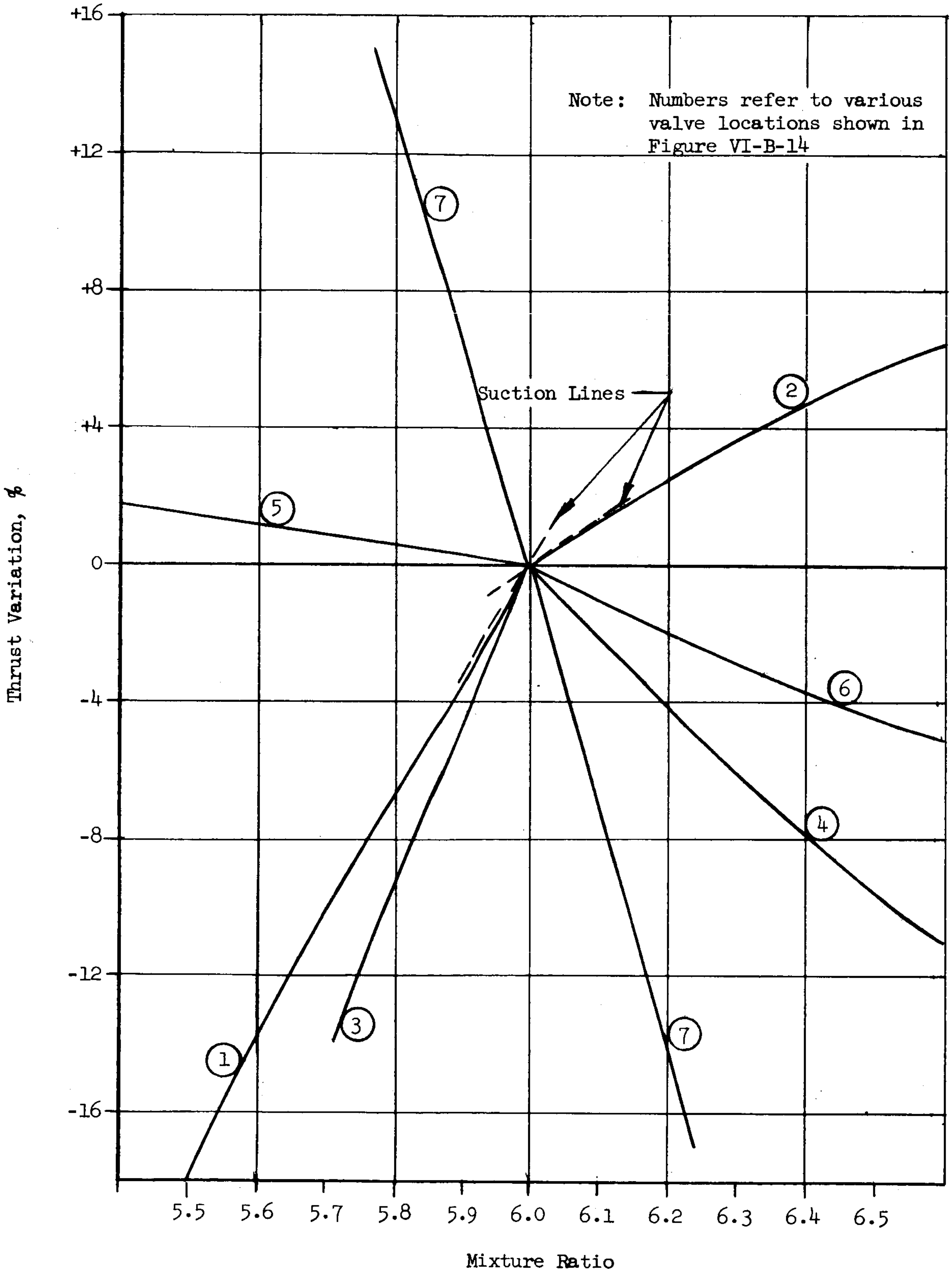
Engine Pressure Distribution

Figure VI-B-13



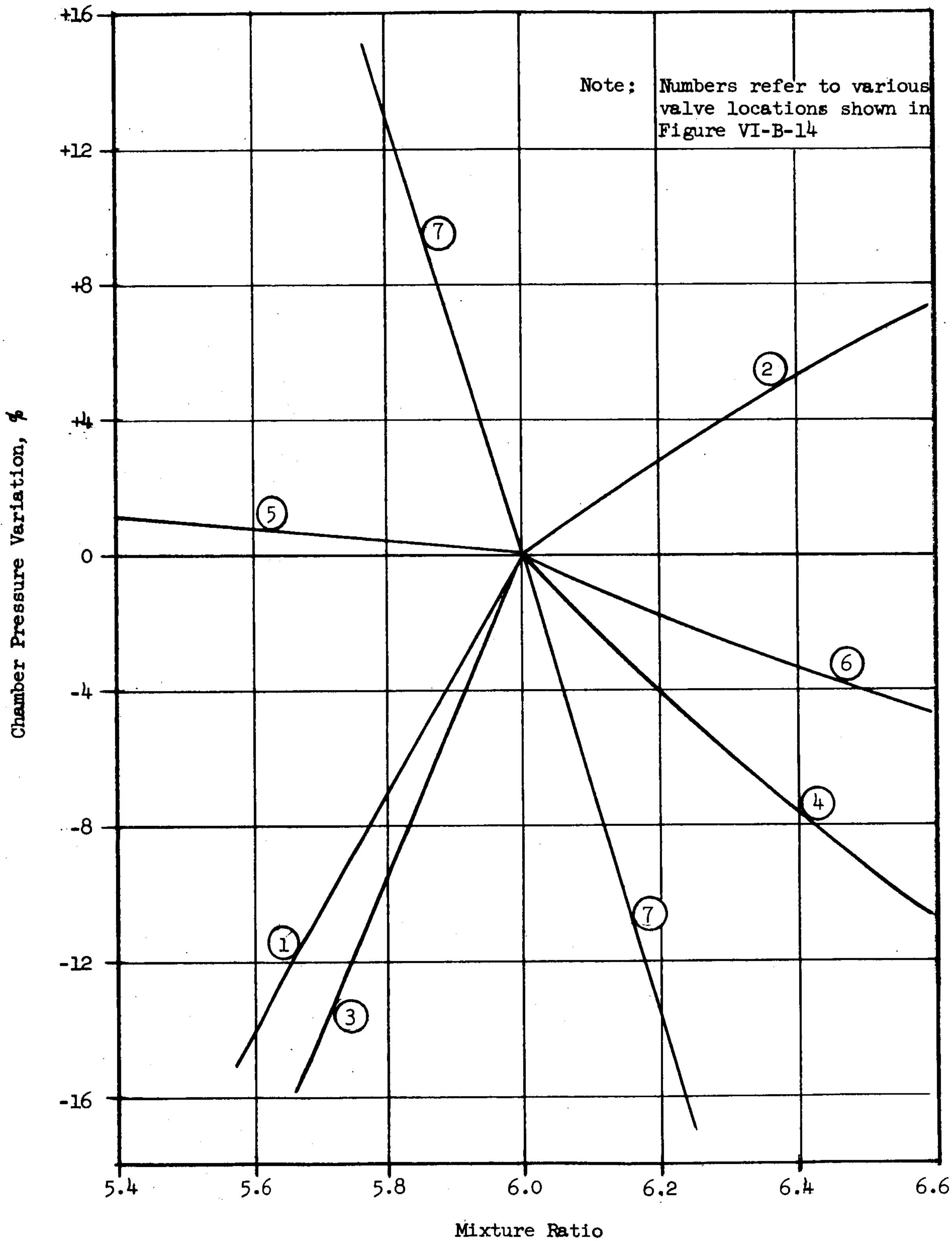
Propellant Utilization, Valve Locations, AJ-1 System

Figure VI-B-14



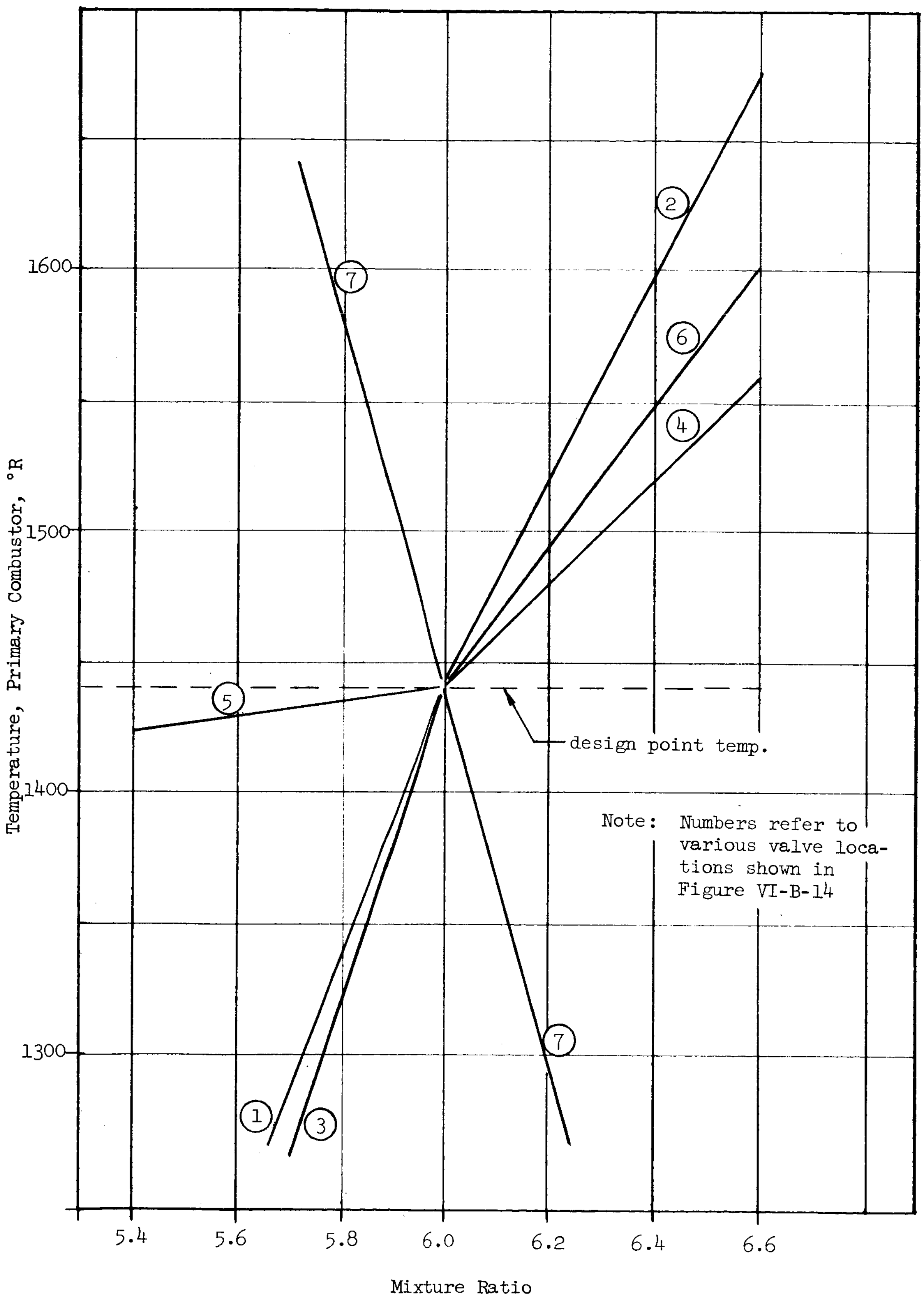
Propellant Utilization, AJ-1 Thrust Variation

Figure VI-B-15



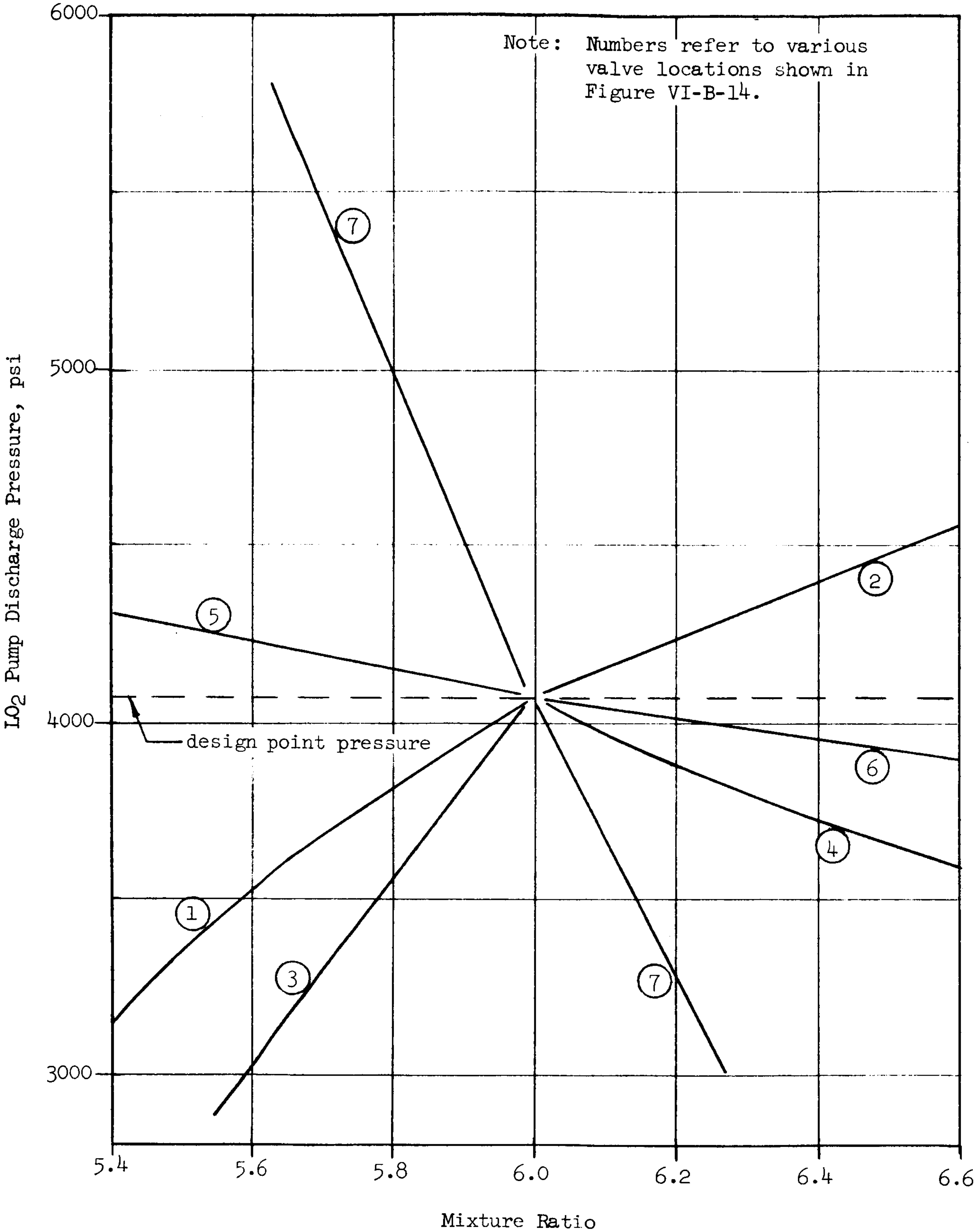
Propellant Utilization, AJ-1 Chamber Pressure Variation

Figure VI-B-16



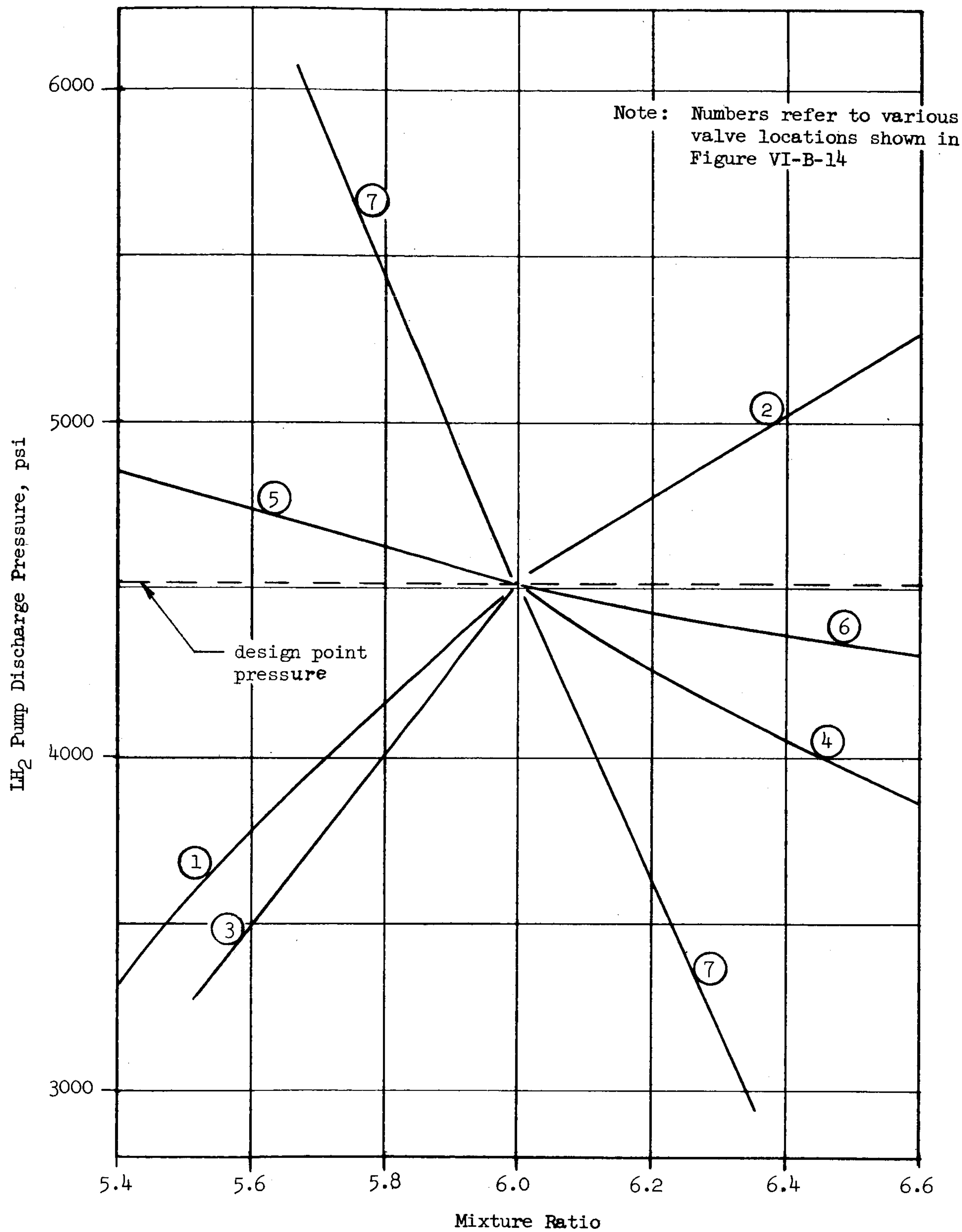
Propellant Utilization, AJ-1 Primary Combustor Temperature Variation

Figure VI-B-17



Propellant Utilization, AJ-1, LO_2 Pump Discharge Pressure Variation

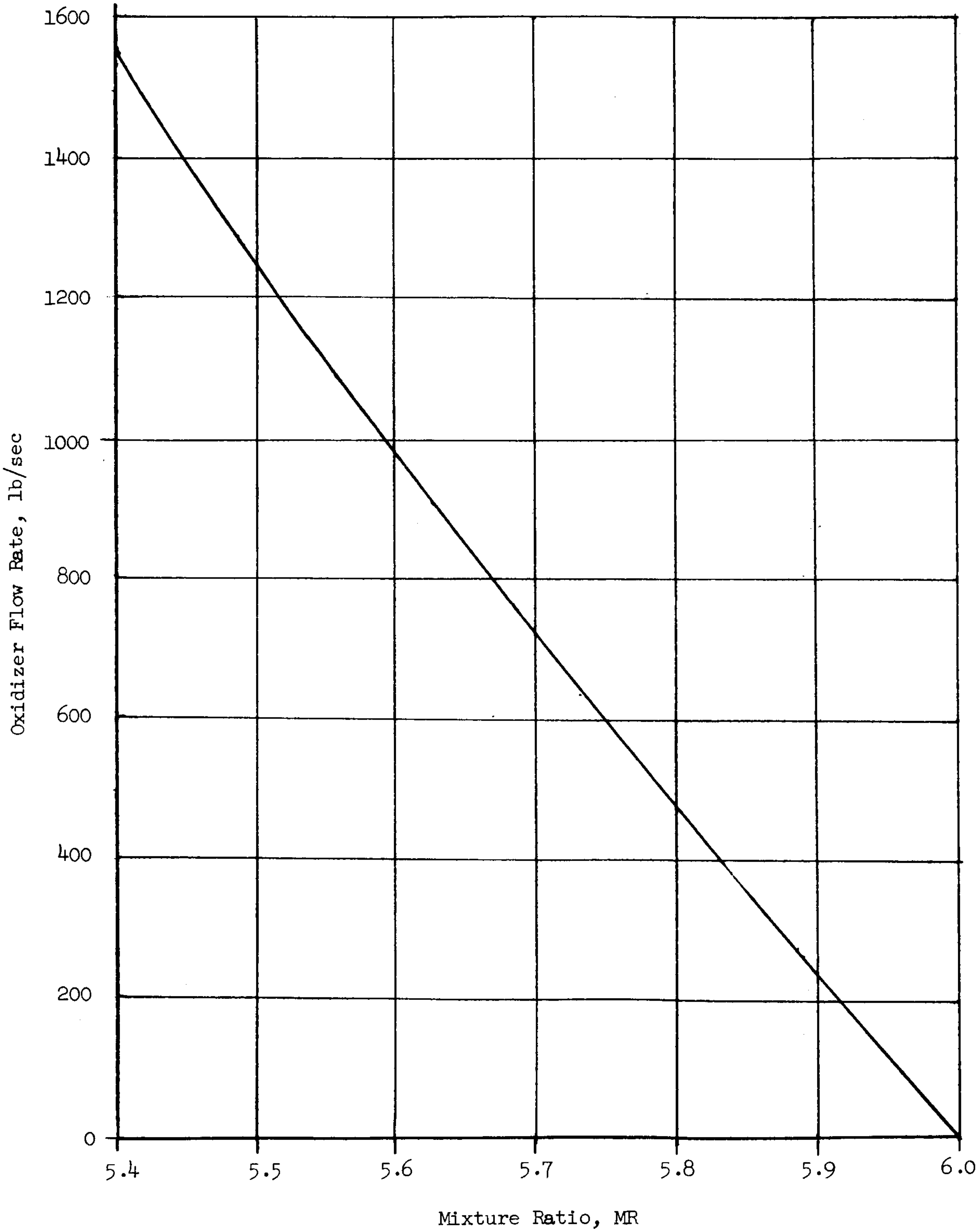
Figure VI-B-18



Propellant Utilization, AJ-1 LH_2 Pump Discharge Pressure Variation

Figure VI-B-19

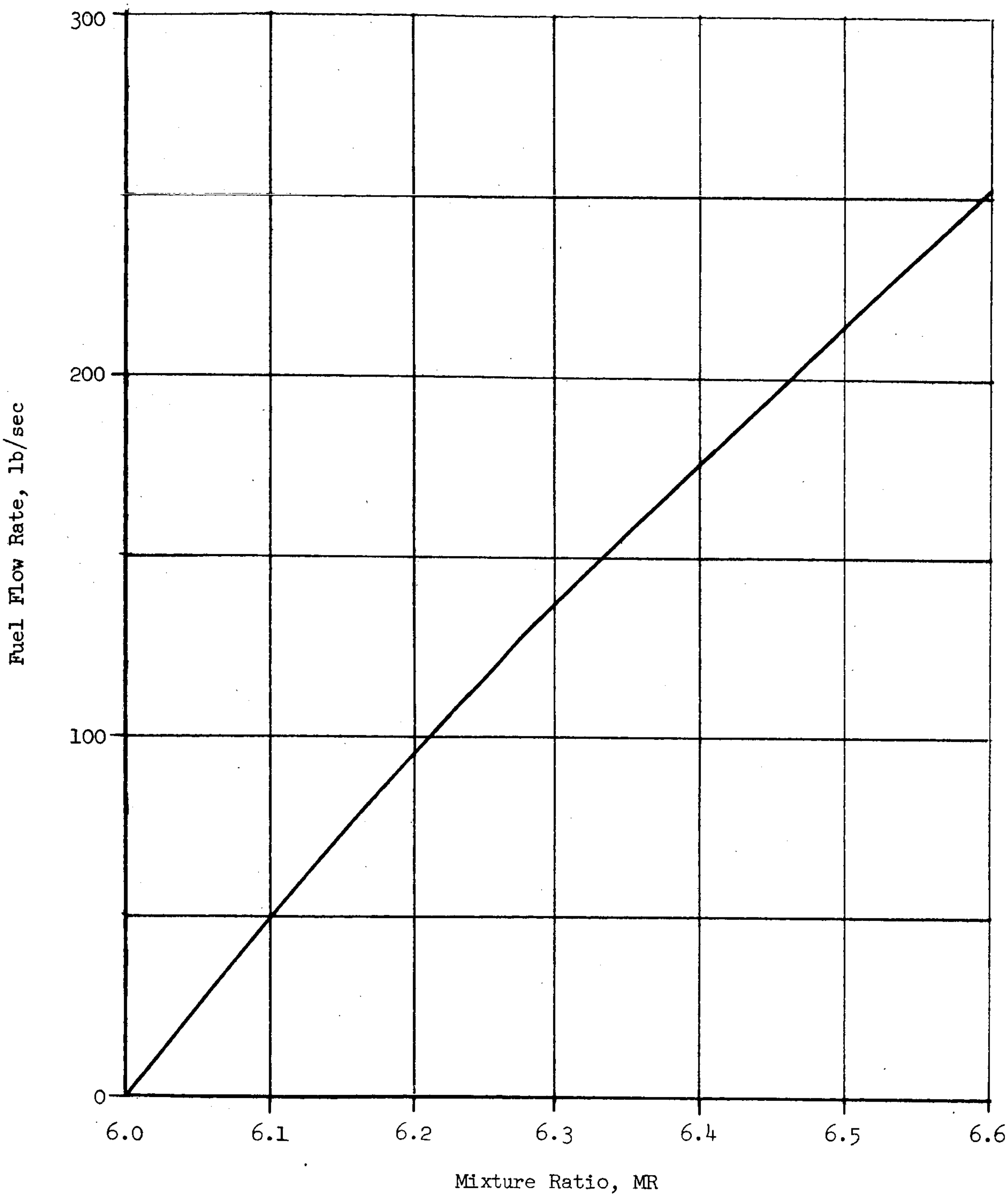
(Second Bypass - Designated 5)



Propellant Utilization, Oxidizer Bypass Flow Rate

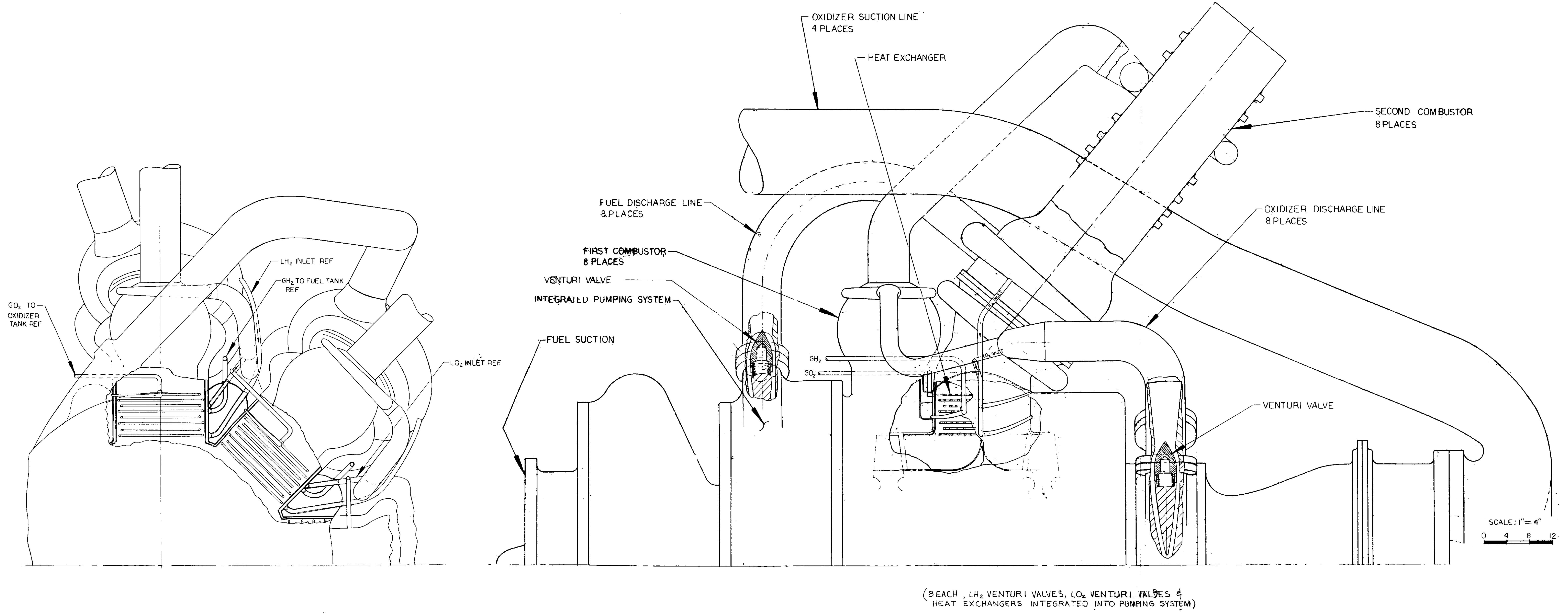
Figure VI-B-20

(Second Bypass - Designated 6)



Propellant Utilization, Fuel Bypass Flow Rate

Figure VI-B-21



AJ-1 Engine with Integral Pump Components

Figure VI-B-22

VI, Engine Systems Studies (cont.)

C. TRANSIENT ENGINE SYSTEM ANALYSIS

1. Start Transient

The objective of performing the transient analysis on this program was to determine (1) valve actuation requirements, (2) possible over-pressures in any component, and (3) special problem areas in any of the engine auxiliary components resulting from the size or pressure of the system.

The transient analysis was performed on Aerojet-General's transient analysis program which is an analytical engine simulation model programed on the IBM 7094 digital computer. The engine simulated by the program usually consists of several component subroutines which, in turn, are represented by a system of differential equations. These differential equations are approximated by numerical difference equations. The integral solution is obtained by solving the difference equation using with a modified Newton-Raphson iteration method and a fixed integration interval throughout each case. Numerical accuracy is specified by input. A predictor subroutine is used to speed convergence. Many parameters are obtained from empirical data; i.e., parabolic-curve-fitted in the form $Y = A + BX + CX^2$; or, the program accepts tabular data in ordered pairs and interpolates internally to obtain a continuous function.

The program considers the engine as a number of individual components, and solves for flow rate, pressure, pressure drop, or any variable desired as a function of time. The order in which the components are solved is at the option of the engineer. The basic input required for the program is as follows:

(1) Line lengths and volumes including manifolds to calculate propellant filling times.

VI, C, Transient Engine System Analysis (cont.)

(2) Pump and turbine performance curves as described in the discussion on steady-state analysis.

(3) Primary and secondary chamber volumes and L*s, (combustion volume/throat area).

(4) Valve resistance versus travel curves for each valve.

(5) Coolant jacket pressure versus volume.

(6) Polar moments of inertia for all rotating machinery.

The program output consists of printout of up to 100 chosen engine parameters at each time point (0.01 sec in this case) and graphs of up to 29 parameters as functions of time.

The study was conducted in the manner described in the following paragraphs.

Once the engine is fully defined, the only variables in the transient analysis are the method of start and the various valve rates associated with that start. It was established in the initial phase of this study that no attempt would be made to optimize an engine transient, and that the method of start would be of secondary importance to the completion of the study. Likewise, maximum pressures and valve timing are relatively insensitive to start method.

VI, C, Transient Engine System Analysis (cont.)

a. AJ-1

An auxiliary gas system start was selected for the major emphasis in this study because this is where the majority of the previous analysis effort centered. A nitrogen start bottle was the choice for the turbine driving force, and valve rates selected were consistent with the state-of-the-art of large valves. Valve rates were selected so the fuel would lead the oxidizer into the primary combustor in order to achieve a smooth rise in combustion temperature up to the design value. (Some LO_2/LH_2 combustion studies have encountered combustion instability when using a fuel lead, but the scope of this program did not warrant trying different sequencing.) Timing was established so that as the fuel cooling-jacket and lines were filled to the primary injector, the oxidizer line and primary manifold were filled. At the time that both systems are filled with propellant and ignition takes place, the turbopump assembly reaches its maximum rotating speed resulting from the nitrogen gas drive. It is at this point that the initial difficulty in the transient occurs. As the primary combustor ignites, the pressure builds up to a value within one or two psia of the fuel pump discharge pressure, and flow essentially stops. The turbopump has insufficient energy at this point to bootstrap itself to the proper speed. For this reason, gaseous helium was substituted for nitrogen to obtain a more energetic start. Figures VI-C-1 and -2 are based on this helium start.

The valve rates were adjusted throughout the study to meet the particular requirements of the current time point. The engine proved to be extremely sensitive to the oxidizer thrust chamber valve rate. Because of the long oxidizer suction line, a valve rate anywhere in the oxidizer system that is too fast will result in a rapid decrease in suction pressure and subsequent cavitation of the oxidizer pump (this takes place when the suction pressure drops to vapor pressure). In addition, if the thrust chamber valve (TCV) rate is too fast, and the flow rate increases sufficiently, the resultant decrease in primary combustor flow results in a decrease of turbine power and speed and the pressure then drops. If the TCV is opened too slowly, primary flows increase and thrust

VI, C, Transient Engine System Analysis (cont.)

chamber pressure does not increase significantly. As this takes place, the turbine tends to overspeed and pump discharge pressures rapidly increase beyond reason. One of the apparent causes of the first problem is the long suction line; however, this is also what keeps the engine operating during primary ignition. The 250-psia spike in suction pressure drives the oxidizer pump discharge pressure sufficiently above the primary chamber pressure to aid the bootstrapping through this critical period. This engine start transient can be developed to the point where all linear valve rates would be used (see discussion of AJ-8, following). The resistance characteristics of the valves can be modified through design to meet this requirement. The valve reaction times required (1 to 3 sec) are not unusual. The various ramifications of these valve actuation requirements are discussed in Section VII of this report. There will be no over-pressures observed except in the oxidizer suction line, as noted above. It is emphasized that all of the problems discussed above become minimal when the transient is optimized. This fact is clearly brought out in the analysis of Engine AJ-8 wherein the results show elimination of these effects.

b. AJ-8

The start transient on the AJ-8 engine was approached in the same manner as on AJ-1. These significant differences exist between the two engine transients: (1) nitrogen was used rather than helium to spin the turbines, and (2) linear valve rates were achieved.

Unlike AJ-1, gaseous nitrogen was sufficiently energetic to start the engine and carry it through primary ignition.

Like AJ-1, nearly linear valve resistance versus stroke characteristics were used. In this case, however, the analysis was optimized and a transient was achieved with linear valve actuation. The valves used were all of the poppet type; the fastest opening time was 0.5 sec for the fuel thrust chamber valve which is opened first to provide coolant flow to the thrust chamber tubes. These valve times together with the pertinent pressures and flow rates are shown in Figures VI-C-3 and -4.

VI, C, Transient Engine System Analysis (cont.)

2. Shutdown Transient

a. AJ-1

The shutdown transient analysis is handled in exactly the same way as the start transient. The philosophy of this shutdown is to start shutting off the main oxidizer supply and throttle the primary combustor while closing the oxidizer thrust chamber valve. The fuel valve is closed last so that both mixture ratios decrease, with the resultant decrease in temperatures, as the engine shuts off. The valve closing times are 0.5, 0.75, and 1.00 sec in the order mentioned above.

The partial results of this shutdown analysis are shown in Figure VI-C-5. The analysis was not completed because difficulties with the start transient delayed initiation of the shutdown analysis. It can be seen from the figure that all pressures rise slightly before dropping off. For the fuel pump, this is caused by the characteristics of the pump itself. That is, it is operating at design conditions slightly to the right of the peak on its head-capacity curve. In the case of the oxidizer pump there are two reasons. First, the operating point on the head-capacity characteristic curve is in a similar location to that of the fuel pump. Second, the pump suction pressure is greatly increased because of the long length of the oxidizer suction line. There is no suction line on the fuel side. Because of the time point at which the analysis ceased, it is impossible to predict what the peak suction or discharge might be for the oxidizer pump; however, the water-hammer analysis in Section VI,C,3 discusses this effect.

b. AJ-8

As in the case of the start transient, the shutdown analysis on AJ-8 was conducted in the same manner as that on AJ-1. In this case, the igniter valves are closed first to stop fuel flow to the oxidizer-rich primary

VI, C, Transient Engine System Analysis (cont.)

combustor (again a lower temperature). Next, the oxidizer valves to both combustors and the fuel valve to the fuel-rich combustor are closed. When the fuel pump discharge pressure decays to less than 650 psia, the fuel thrust chamber valve is closed which stops the coolant flow. All valves are closed in 0.5 sec, except the thrust chamber valve which closes in 2 sec. In this shutdown the only over-pressures observed in the system are in the suction lines: 66 psia in the oxidizer line and 50 psia in the fuel line. Figure VI-C-6 summarizes these shutdown results in graphical form.

In considering all of the results presented from the engine transient analysis, it can be seen that the valve rates are not unusual and that there will be no unique actuation requirements due to the size or pressure of the system. Any over-pressures in the engine can be eliminated by optimizing the valve rates. There are no special problem areas in any of the engine auxiliary components attributable to the start or shutdown of the engine system.

3. Water-Hammer Effects

Calculations were made to determine the peak pressures that could result from water-hammer effects in the AJ-1 engine system. Water-hammer is the phenomenon of pressure waves in a system of conduits which occur when the velocity of the fluid is suddenly changed. Any such change, if sudden enough (as when a valve is opened or closed), can result in damaging pressure spikes. Consideration of water-hammer is very important when designing auxiliary components; therefore, a treatment of the phenomena is included in this investigation.

VI, C, Transient Engine System Analysis (cont.)

The fundamentals of water-hammer, based on the elastic column therory, can be summed up in three basic equations. They are:

$$\Delta H = \frac{a \Delta V}{g} \qquad \text{(Eq 1)}$$

$$a = \frac{\sqrt{\frac{K'g}{\rho}}}{\sqrt{1 + \frac{K'D}{Et}}} \qquad \text{(Eq 2)}$$

$$T = \frac{2L}{a} \qquad \text{(Eq 3)}$$

where ΔH = Change in head due to pressure wave, ft

a = Acoustic velocity, ft/sec

T = Time for wave to return to valve, sec

V = Velocity change of fluid in pipe, ft/sec

K' = Bulk modulus of fluid, lb/in^2

D = Diameter of pipe, in.

ρ = Density of fluid, lb/ft^3

t = Thickness of pipe, in.

L = Length of pipe, ft

g = Gravitational constant, 32.17 ft/sec^2

E = Modulus of elasticity of pipe wall, $lb/in.^2$

Equation 1 determines the water-hammer pressure rise in the pipe for instantaneous valve closure; that is, when closure is so fast that rarefaction (reflected) waves cannot return to the valve before closure is complete. Equation 2 determines the velocity of the pressure waves (acoustic velocity) that originate at the valve the instant closure begins. Equation 3 determines the time required for the pressure wave to traverse the line from the valve to some reflecting element (such as a tank) and back again. It must be emphasized that rarefaction waves have a great influence on the peak pressures created at the valve. If closure is gradual enough so that interaction between direct and rarefaction waves occurs, the calculation of peak water-hammer pressure becomes more complicated than by simply employing Eq 1. For instance, anything that causes an increase in the acoustic velocity (a) will by Eq 1 cause an increase in the pressure rise (ΔH). However, an increase in (a) also means that the wave (according to Eq 3) will travel through the line much faster. This, in turn, means that unless valve closure time is shorter than $2\frac{L}{a}$, rarefaction waves will return and reduce the pressure waves originating at the valve. Thus, for gradual valve closure, there is an interplay of these two opposing phenomena. The analytical relationship between (T) and (ΔH) is complicated, and the solution is normally determined by graphical methods or with a computer into which the graphical method is programed.

The problem was further compounded by the complexity of the AJ-1 system. A schematic of the engine is shown in Figure VI-C-7. The sketch represents one of the four suction lines and a branched network containing two of the eight secondary combustors. The tank was assumed to be a perfect reflector of the pressure waves originating at the valve. Shown also are a number of parameters necessary for solving the equations. Although other fluids were examined, liquid oxygen was used most extensively throughout the study. Hence, the indicated fluid parameters are for LO_2. The computer program required that pressure drops be lumped at various points along the line. In addition, the program required the chamber pressure to remain constant during valve closure--an assumption that is valid when valve closure occurs in the suction line, but faulty when closure occurs in the pump discharge line. The rarefaction waves were assumed to be of the same magnitude as the direct waves but opposite in sign. Only the shutdown transient was considered, and valve closure in most cases was performed in the suction line.

a. Valve Types

Five different types of valve resistance curves, i.e., K-factor versus stroke, were studied to determine their influence on water-hammer pressures.

VI, C, Transient Engine System Analysis (cont.)

These are shown in Figure VI-C-8. The value of K at the full-open position was designated to be the same for all types, and was calculated from the following equations:

$$\Delta P = 1.4 \frac{\mathring{w}^2}{\rho D^4}$$

$$K = \frac{\mathring{w}}{\sqrt{\Delta P}}$$

where ΔP = Pressure drop across valve, psi

K = Valve resistance factor, lb/sec $\sqrt{psi}$

$\mathring{w}$ = Fluid flow, lb/sec

ρ = Fluid density, lb/ft^3

D = Valve diameter, in.

Each resistance curve was evaluated at three closing times: 0.25, 0.50, and 1.00 sec. Liquid oxygen was the fluid. The shutdown valve was placed in the suction line just forward of the pump.

The results are shown in Figure VI-C-9, which is a plot of maximum total pressure encountered at the shutdown valve versus the time to close the valve. Each of the five curve types is indicated.

The lowest pressure peaks occurred with Types 1 and 5, whereas Types 2 and 4 created pressure peaks very near that value calculated for instantaneous closure. Type 3 created pressures between the two sets. Actually, the pressures for Types 2 and 4 were not too severe until the very last 0.03 sec before complete closure. A close examination of the computer data revealed that, at this point, very sharp pressure spikes occurred. These spikes were a result of very rapid increases in the fluid velocities, which in turn were caused by the sharp decrease in the K values near zero stroke (Figure VI-C-8). Thus, initial valve

VI, C, Transient Engine System Analysis (cont.)

closure was slow and the flow rate was still relatively large when the valve was only 10% open. At this point the valve was essentially slammed shut. This last phase resulted in the creation of large pressure waves, but the valve was closed by the time the corresponding rarefaction waves returned. In effect, overall closure was nearly instantaneous. These data and those following demonstrate the importance of rarefaction waves.

b. Valve Location

To find the valve location resulting in the least pressure rise, valve Type 1 was also placed in various points along the suction line. Using LO_2 and the same closing times, the maximum pressures were again calculated, this time with the results shown in Figure VI-C-10.

The maximum peak pressures occurred with the valve in the center of the line. When calculations are based on the rigid column theory, maximum pressures would occur with the valve moving downstream from the tank.

The results, however, are reasonable if one considers that the pump acts as a constant pressure reservoir similar to the tank. This is possible because almost immediately after closure begins, cavitation occurs just downstream of the valve resulting in a pressure equal to the vapor pressure of the fluid. Therefore, when the valve is placed near the pump (or the tank) rarefaction waves return almost instantly to the valve and reduce the pressure. When placed at the center of the long suction line, the rarefaction waves return somewhat later and allow a greater pressure rise before interaction can occur.

The preceding areas, i.e., valve types and location, comprise the more pertinent parts of the investigation. However, during the course of the study, the shutdown valve was also placed in the pump discharge line. The valve having the same shape resistance curve as Type 1 was closed in 0.50 sec. The fluid was again LO_2, and the pump discharge was 4100 psi.

VI, C, Transient Engine System Analysis (cont.)

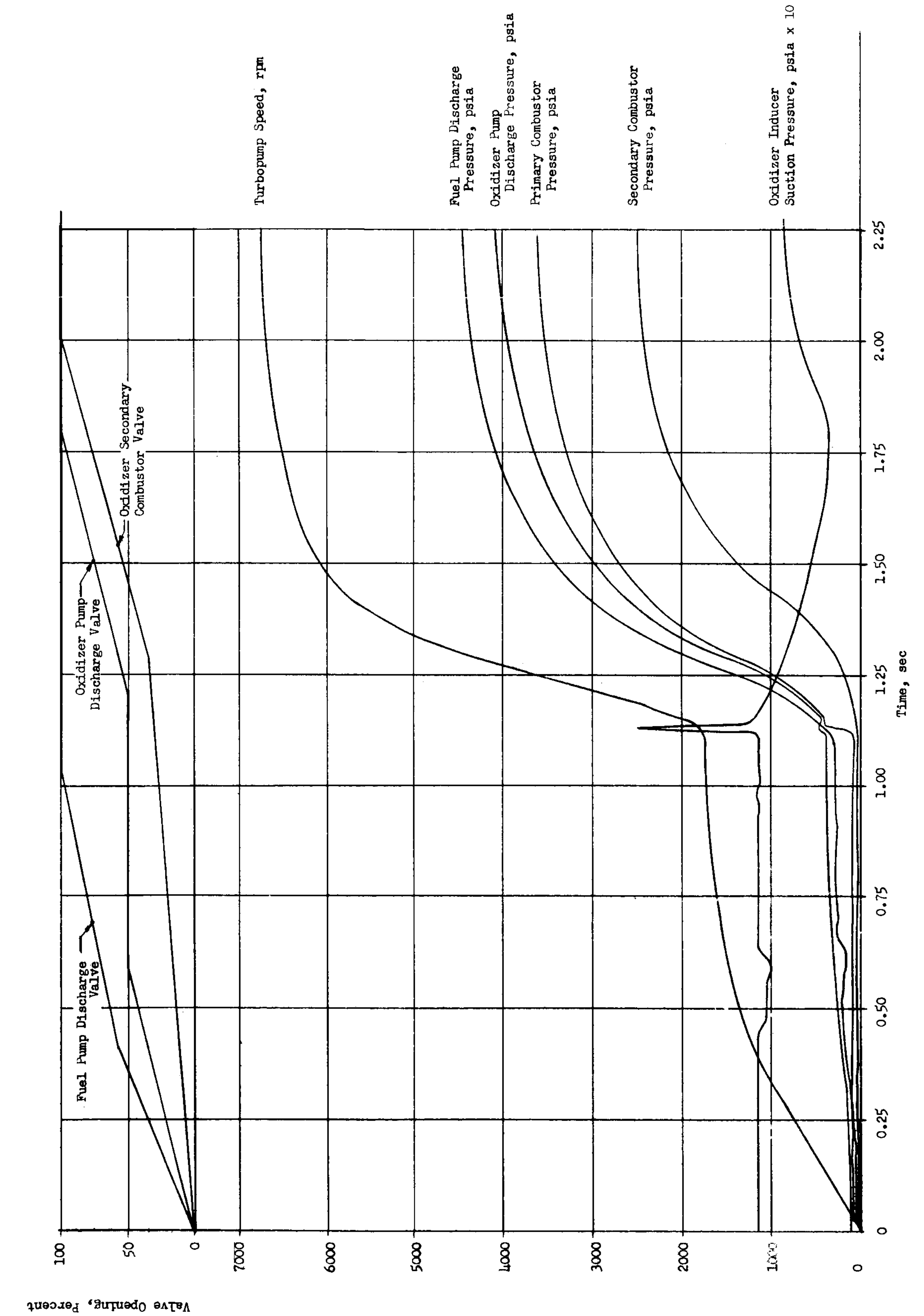
The results in Figure VI-C-11 show the increase in pressure to be about 300 psi in the discharge line and about 40 psi in the suction line. The former is about 250 psi greater than when the valve is closed in the suction line next to the pump.

A few computer runs were also made to compare RP-1 with LO_2. The fluid velocities were held the same and the Type 1 valve was again placed in the suction line, next to the pump. The closing times were 0.25, 0.50, and 1.00 sec.

The results are shown in Figure VI-C-12. It is a plot of maximum total pressure in the suction line as a function of valve closing time. The latter is given in terms of its ratio to the instantaneous closure time, $2\frac{L}{a}$. Shown also are the various values of densities and bulk moduli for the fluids, including those for LH_2. For all closure times except instantaneous closure (abscissa equals 1.0), RP-1 gives greater water-hammer pressures than LO_2. The reverse is true at instantaneous closure. The curve for LH_2 is what would be expected, based on a calculated pressure for instantaneous closure.

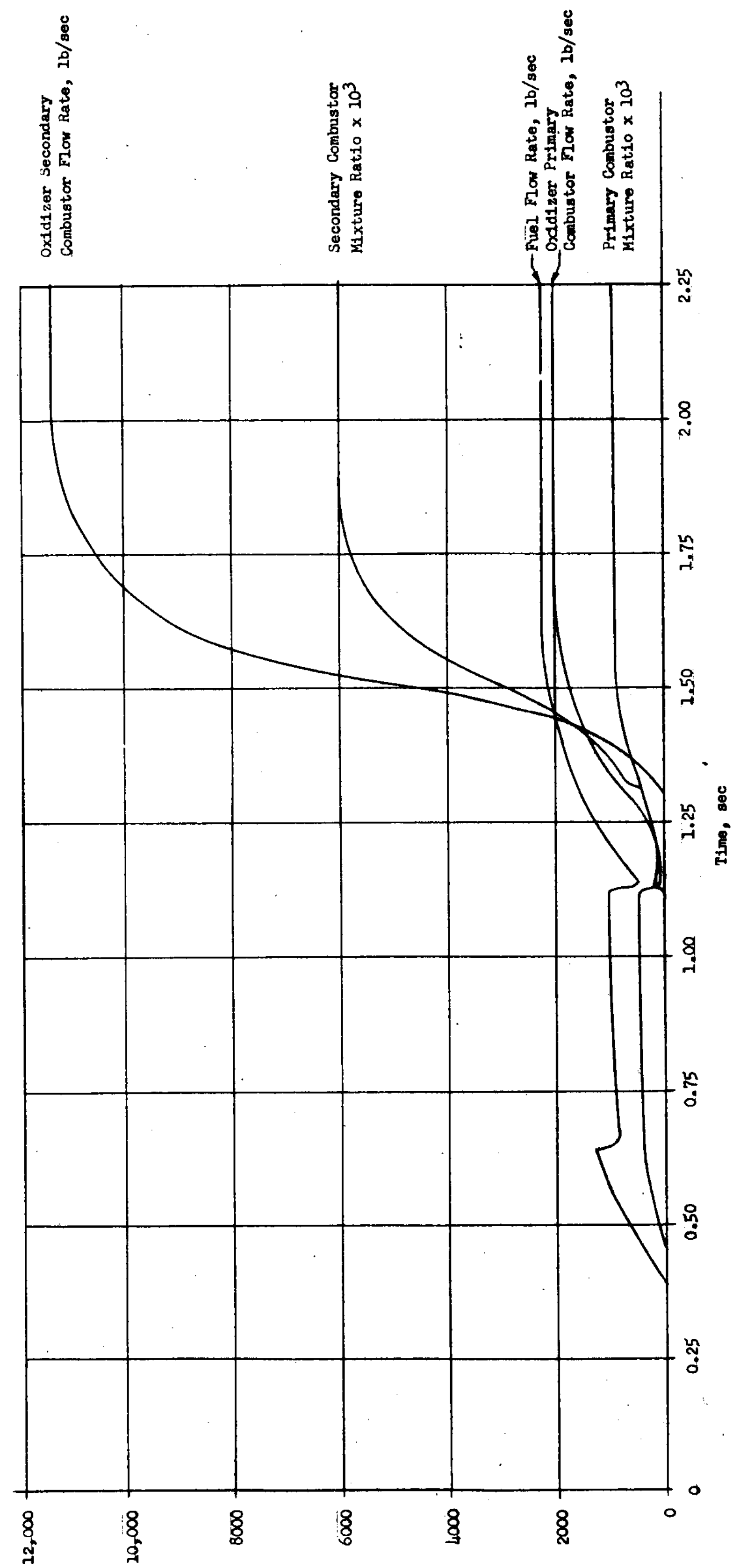
In summary, the results certainly indicate the importance of choosing a proper type of shutdown valve. It should be of a type that produces increasingly smaller changes in fluid velocity, just preceding complete closure. This is one reason that some sleeve and poppet valves are designed for "high resolution at low flow" or equal percentage flow modulating characteristics.

It is preferable to place the valve as close to a reservoir as possible since the highest pressures occur at the center of the line. This requirement becomes increasingly critical as the line length increases.



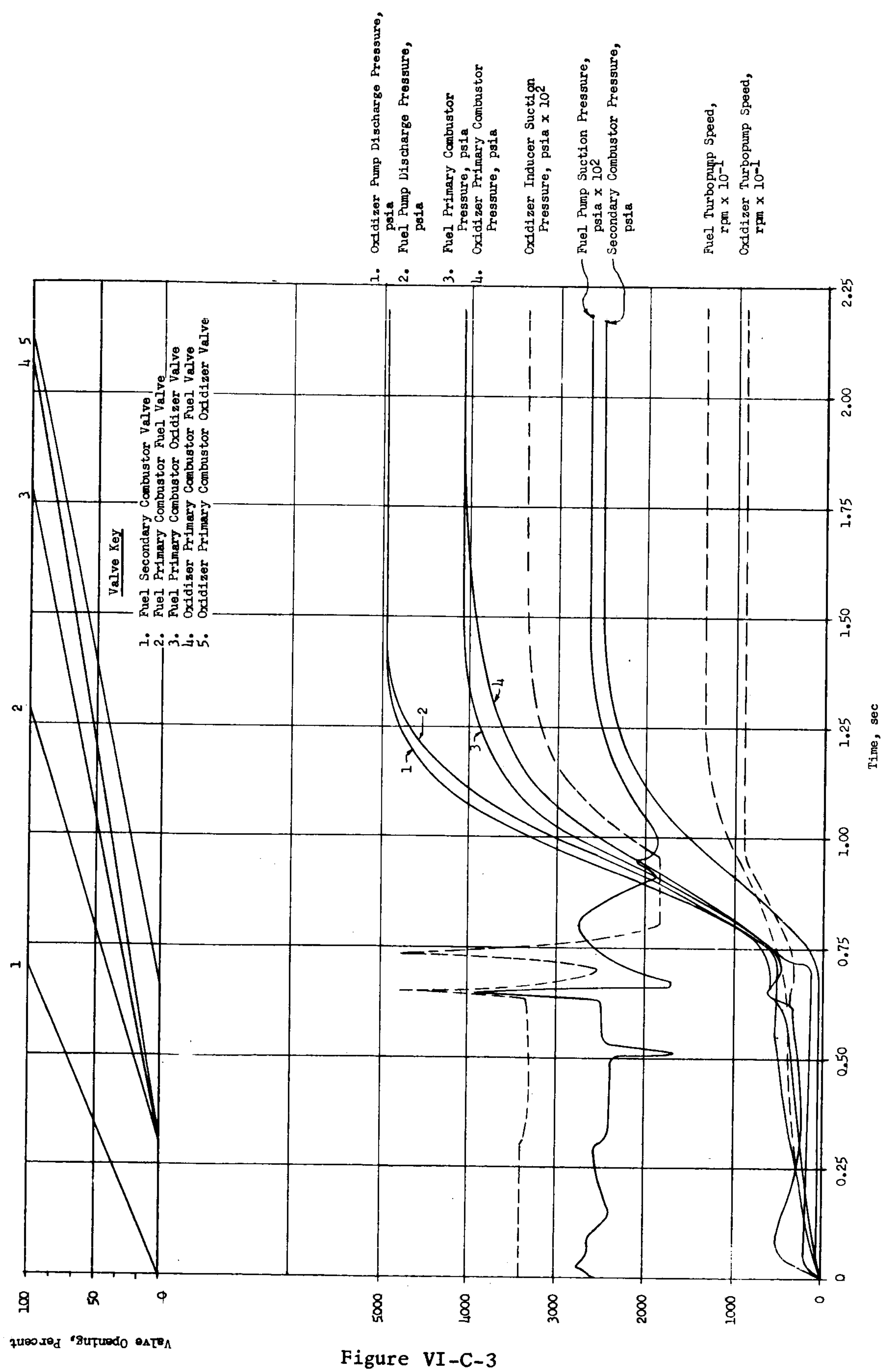
AJ-1 Engine Start Transient, Sheet 1

Figure VI-C-1



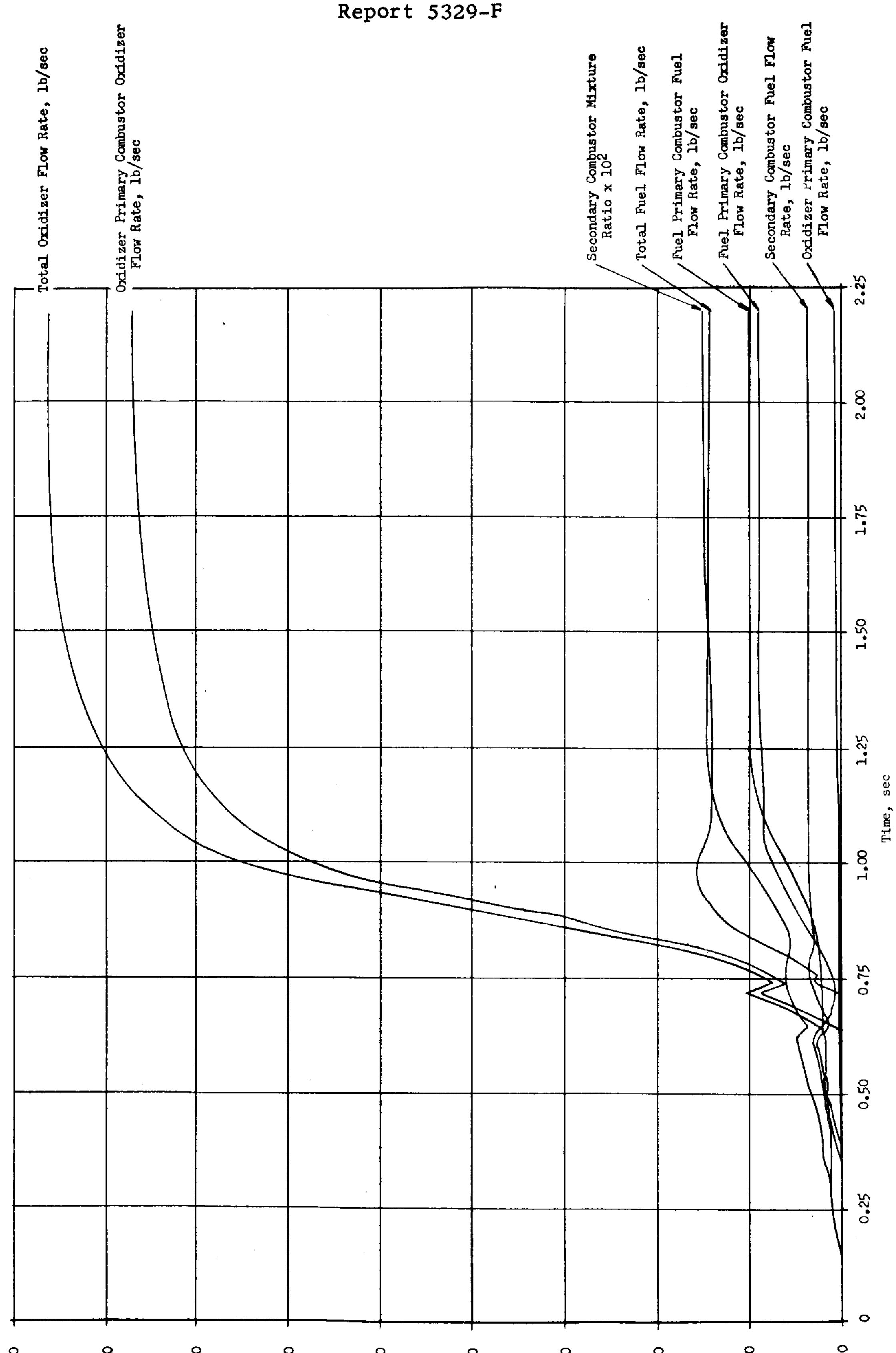
AJ-1 Engine Start Transient, Sheet 2

Figure VI-C-2



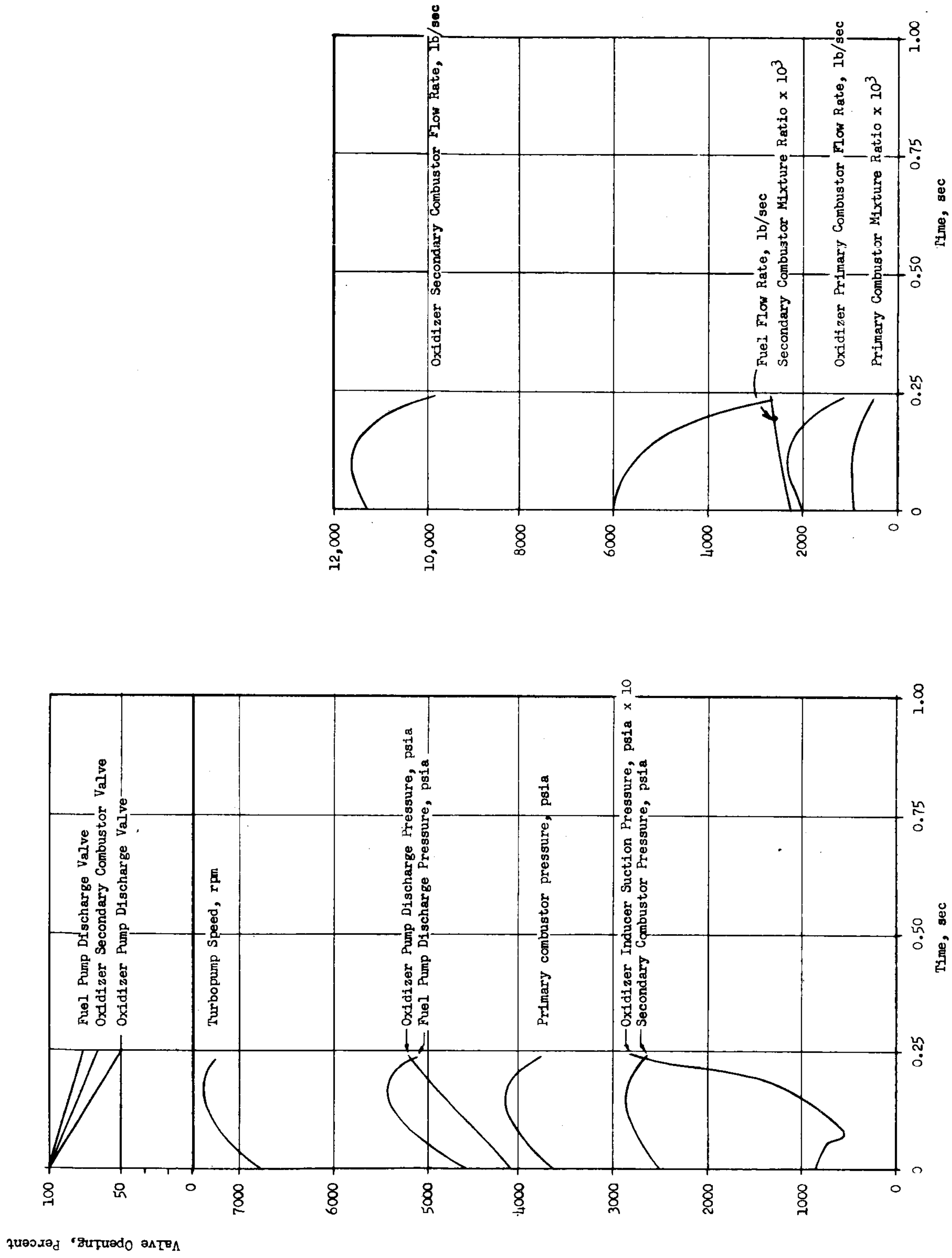
AJ-8 Engine Start Transient, Sheet 1

Figure VI-C-3



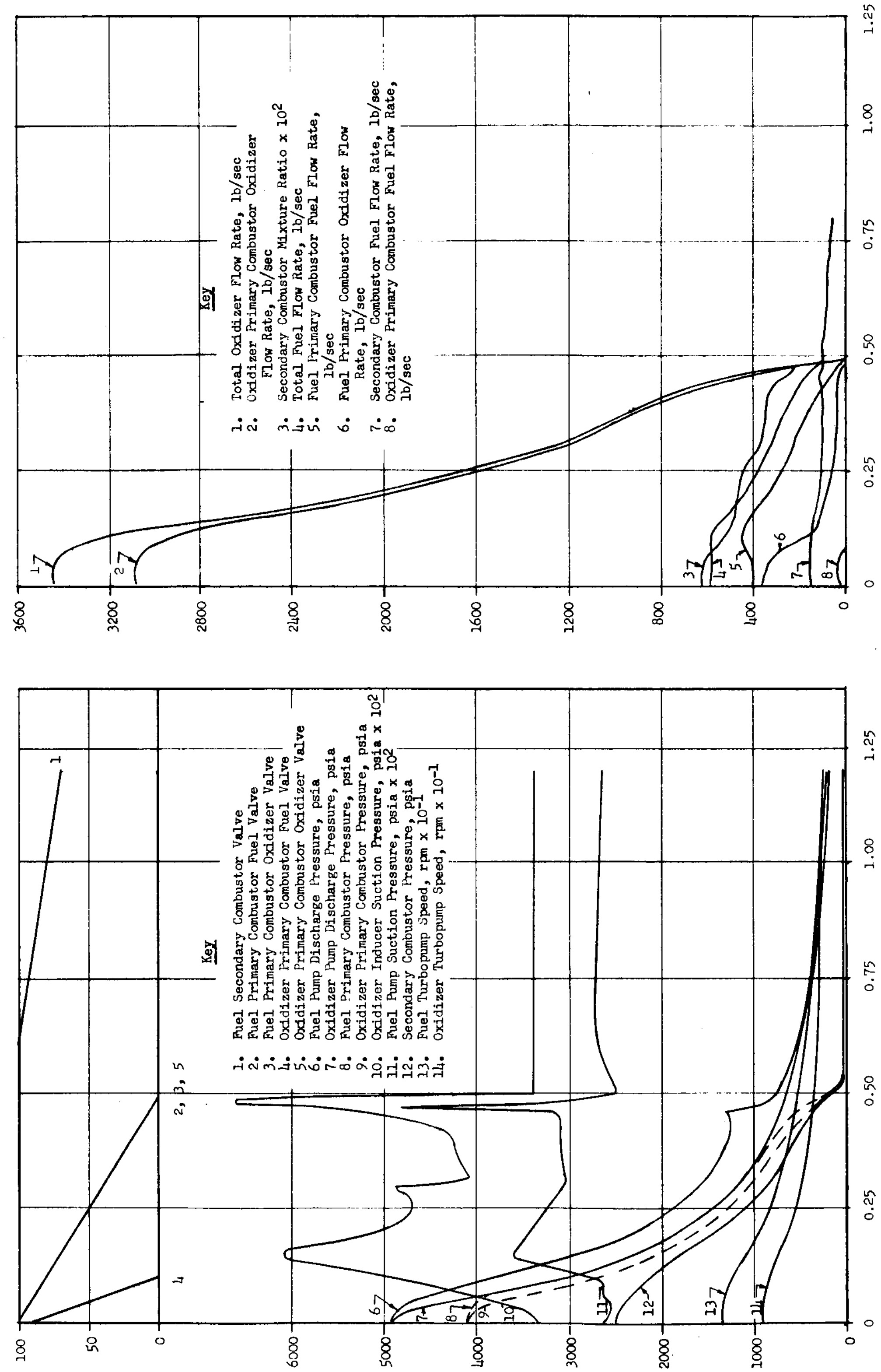
AJ-8 Engine Start Transient, Sheet 2

Figure VI-C-4



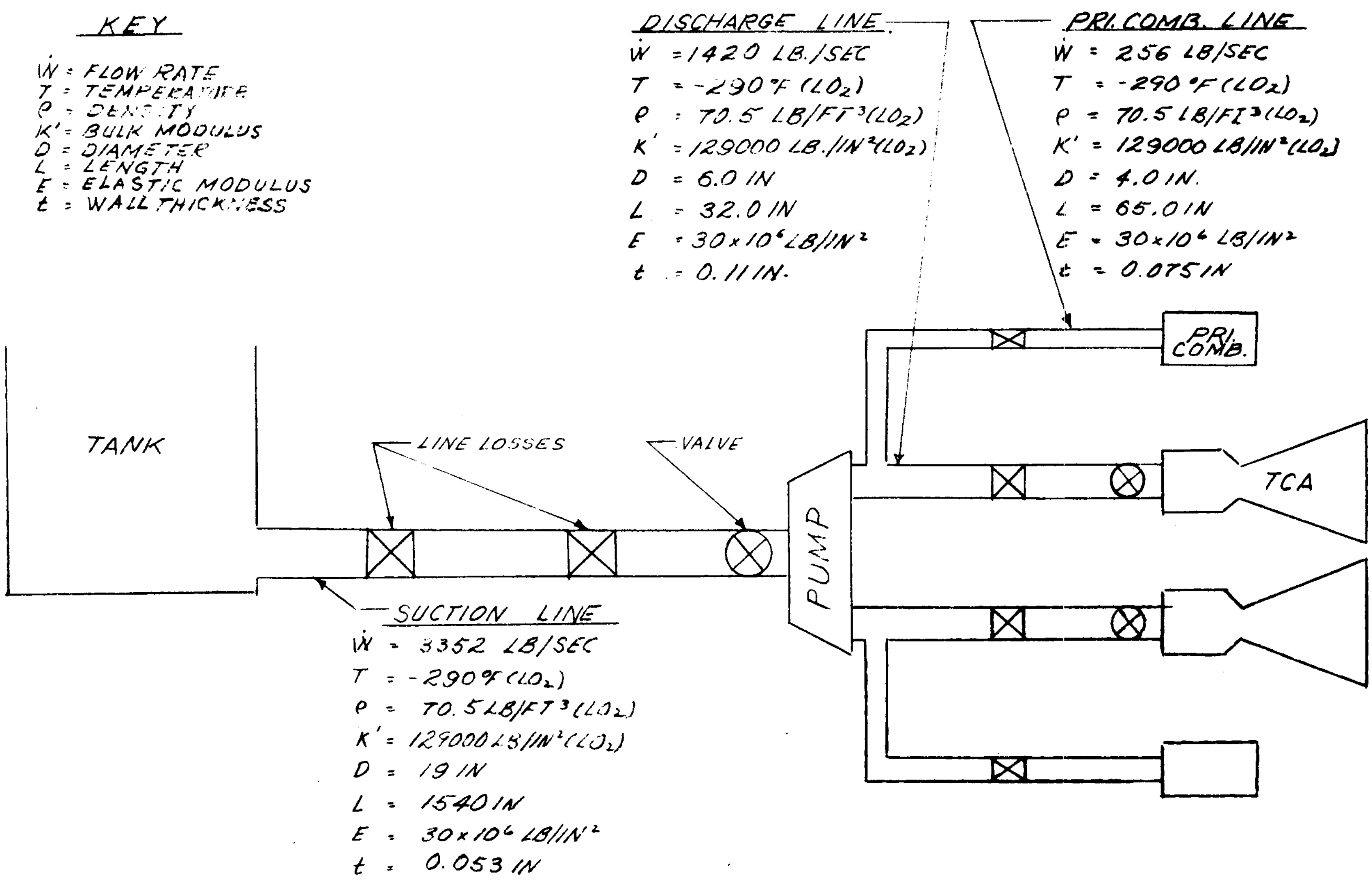
AJ-1 Engine Shutdown Transient

Figure VI-C-5



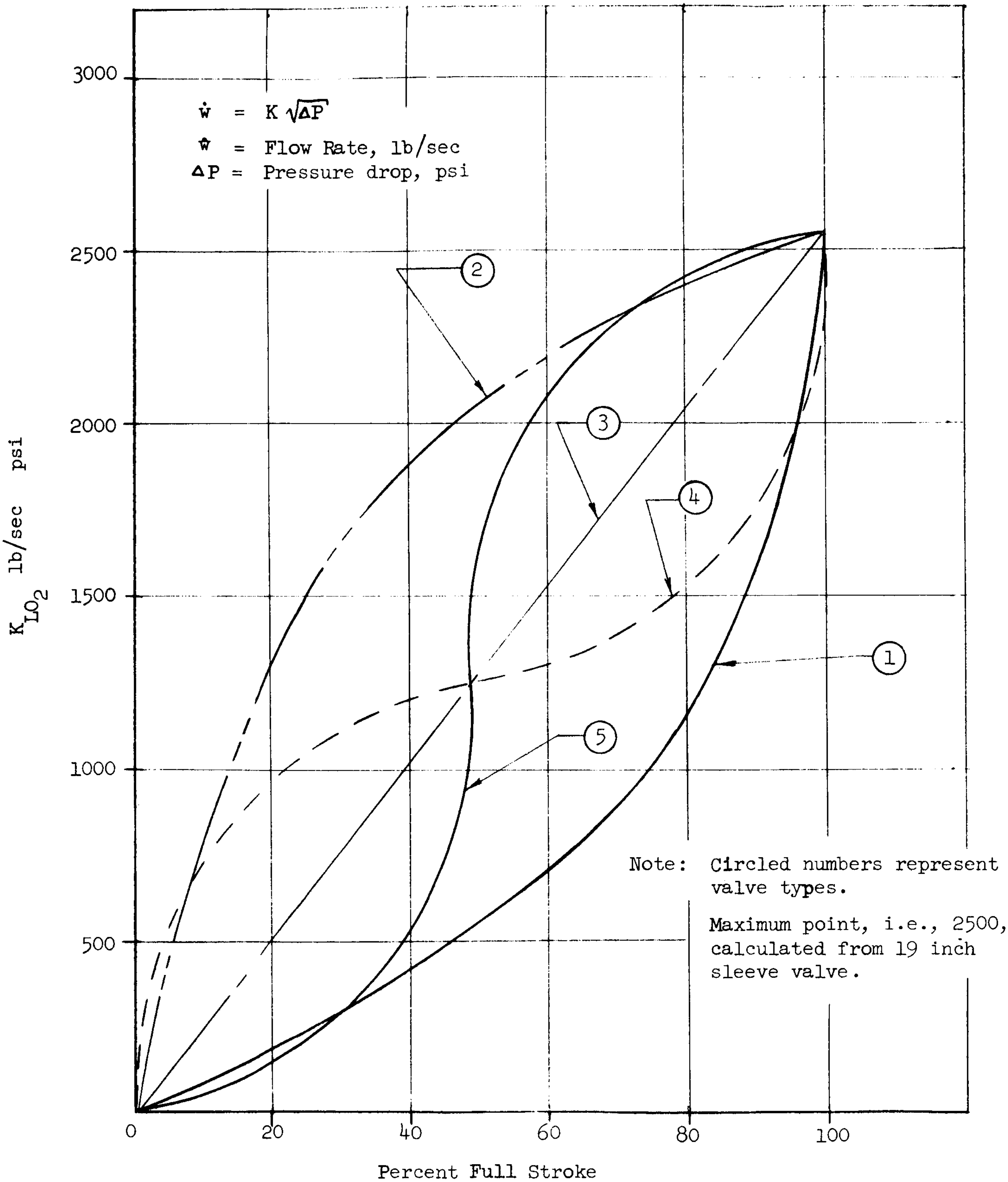
AJ-8 Engine Shutdown Transient

Figure VI-C-6



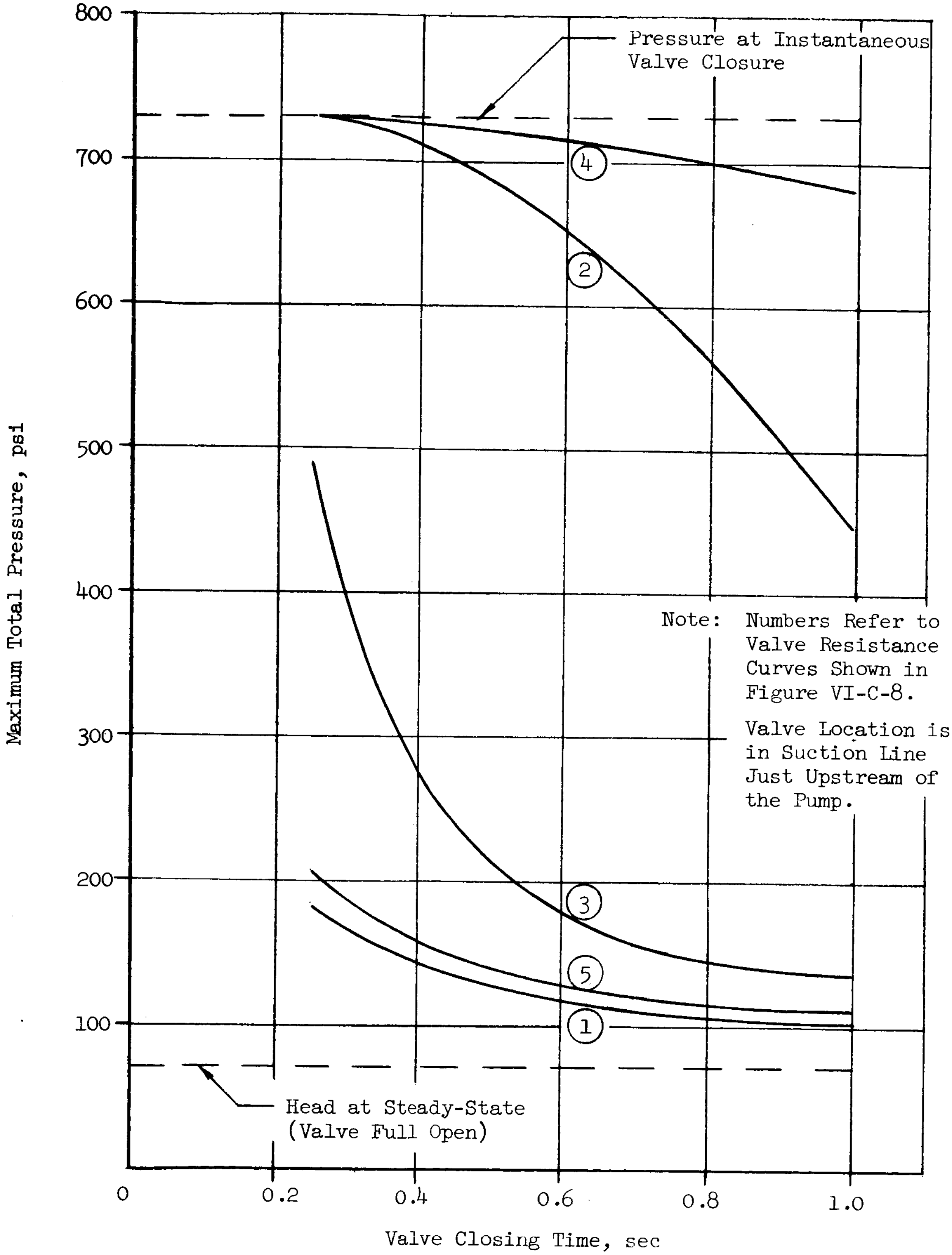
Water Hammer System Diagram

Figure VI-C-7



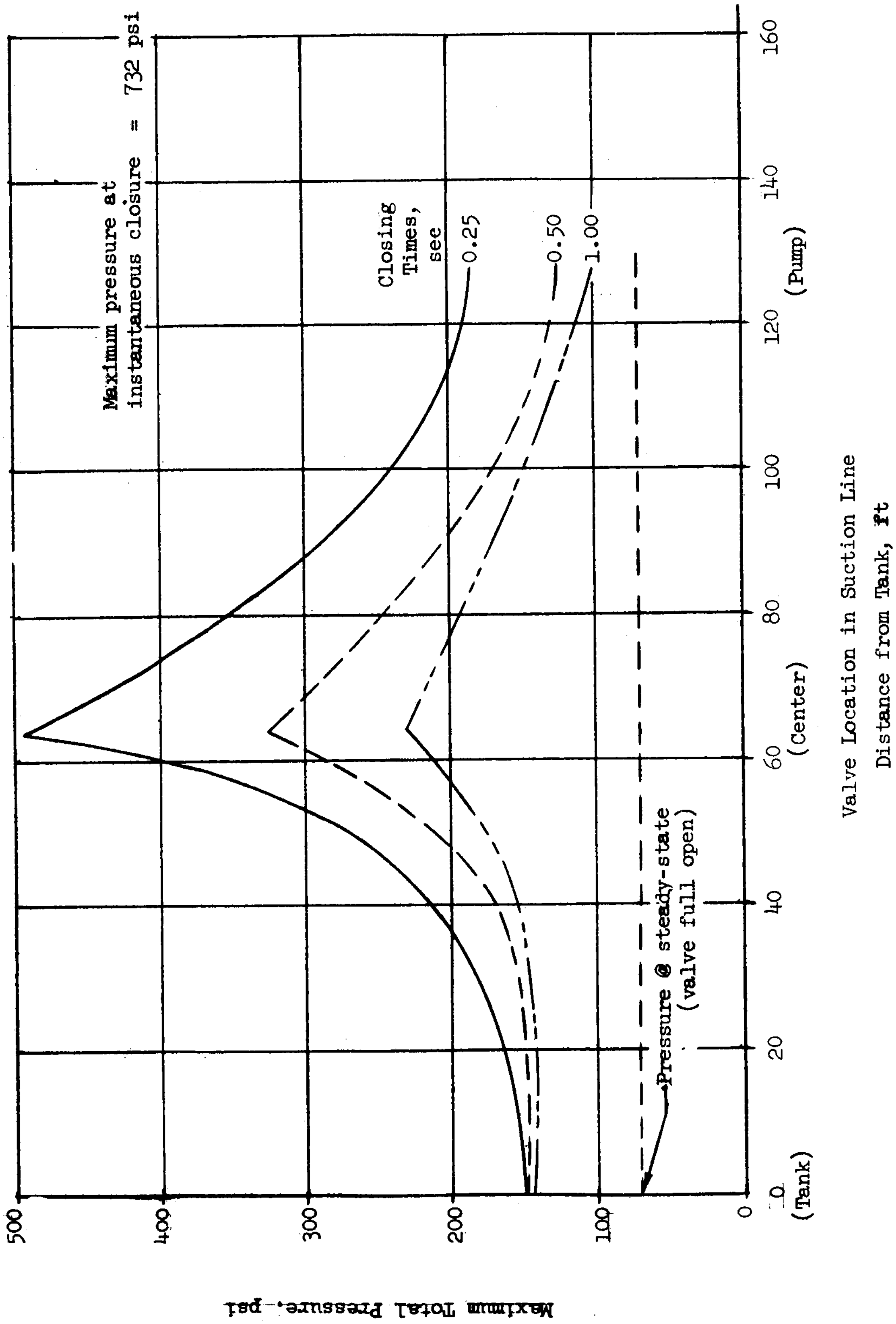
Water Hammer Valve Resistance Curves

Figure VI-C-8



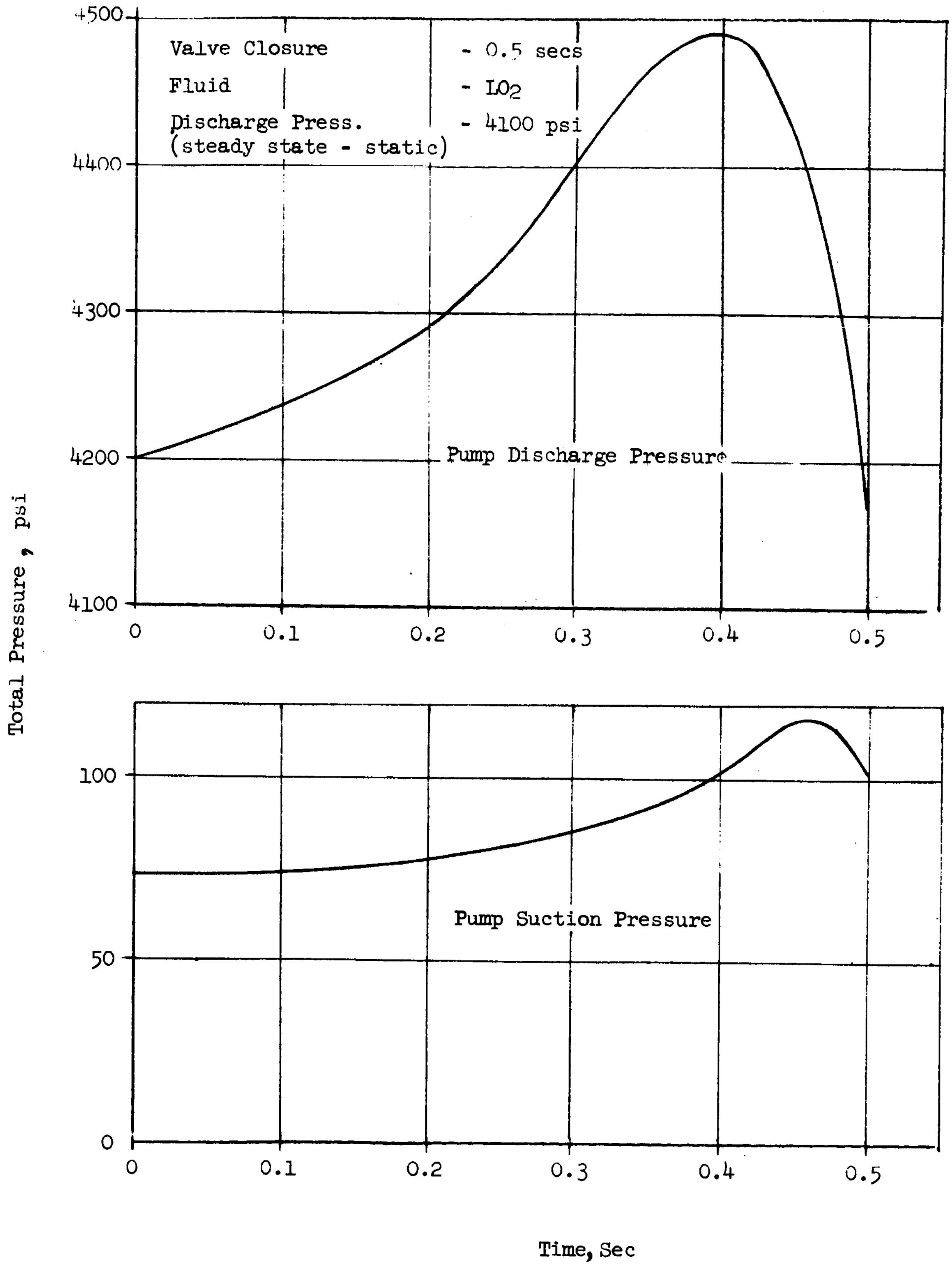
Water Hammer, Maximum Total Pressure as a Function of Valve Closure Time for Various Valve Resistance Curves

Figure VI-C-9



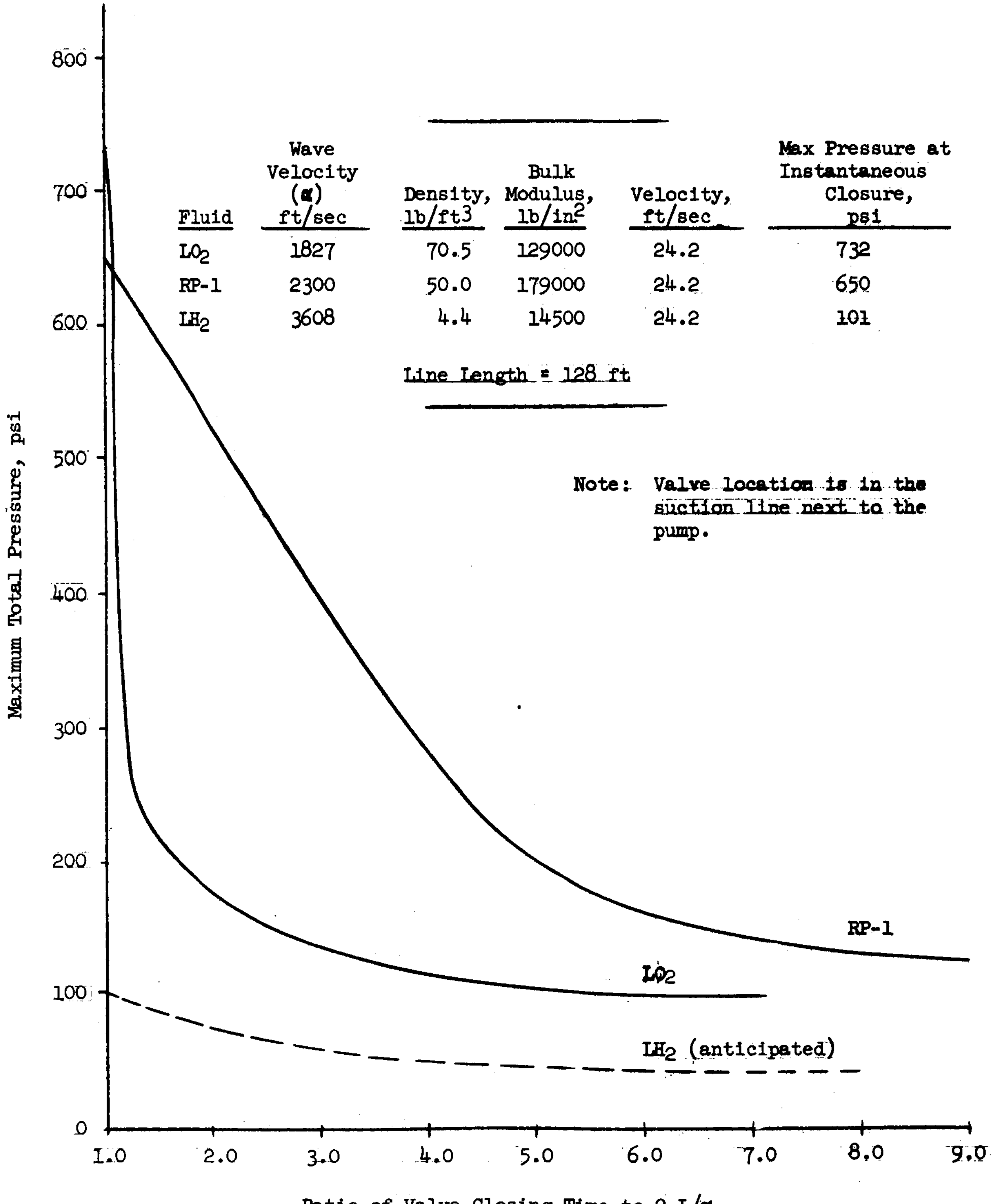
Water Hammer, Maximum Total Pressure as a Function of Valve Distance from Fluid Tank

Figure VI-C-10



Water Hammer, Pressure-Time History for Shutdown Valve in Discharge Line of the AJ-1

Figure VI-C-11



Fluid	Wave Velocity (α) ft/sec	Density, lb/ft³	Bulk Modulus, lb/in²	Velocity, ft/sec	Max Pressure at Instantaneous Closure, psi
LO_2	1827	70.5	129000	24.2	732
RP-1	2300	50.0	179000	24.2	650
LH_2	3608	4.4	14500	24.2	101

Line Length = 128 ft

Water Hammer, Maximum Total Pressure as a Function of Valve Closing Time for Various Fluids

Figure VI-C-12

VII. VALVES

A. METHOD OF ESTABLISHING DESIGN CRITERIA

The objective of the valve studies conducted for this program was to determine the criteria by which the best type of valve can be chosen for a particular set of design requirements for large high-pressure engines, and to evaluate the problems and required technology associated with the design of such valve types. The requirements imposed on the valve concepts can be divided into two broad categories: (1) the basic engine design parameters, such as system pressure, flow rate, and fluid media (propellant); and (2) the engine operating requirements and environment, which include such considerations as flow control (modulating or on-off), resistance versus stroke characteristic, pressure loss, response time, fail-safe provision, allowable leakage, method of actuation, and protection from external environment.

No single valve concept will fulfill all the requirements listed above for all the possible combinations of engine specifications. For example, the ball valve, which has excellent pressure drop characteristics, does not lend itself to propellant actuation by a self-contained actuator. Therefore, many valve concepts were investigated to determine the applicability of each type of valve to the various functions required by the specific engine designs considered in this study program. The results of this investigation will therefore provide a basis by which the best valve concept can be determined for a specific engine valve function. The actual choice of a particular valve concept over all others must necessarily be a process of determining which concept fulfills the greatest number of important requirements while sacrificing to the least extent other desirable features such as design simplicity, weight, ease of fabrication, cost, accessibility to moving parts, etc. The valve ratings given in the summary of design criteria (Table II-1) were based solely on the application requirements and not upon these latter considerations.

A number of approaches were utilized to provide both absolute and comparative data for valve concept comparisons in terms of the parameters mentioned

VII, A, Method of Establishing Design Criteria (cont.)

previously. In the investigation of the different valve concepts, the basic problem of sealing both fixed and moving parts was not considered a limiting factor in determining the feasibility of any valve concept with regard to size, clearances, tolerances, or pressure. To limit the consideration of valves to those which could use present seal technology would necessarily exclude from consideration several concepts that would otherwise be attractive. Since the capability of sealing effectively is of primary concern in nearly all valve functions, the relative magnitude and general nature of this problem is discussed in detail for each of the valve concepts considered. In these studies, the valves are grouped into the following four categories: (1) high-pressure, large flow-rate valves for location in propellant lines (discrete valves), (2) high-pressure valves built into the pump (discharge) housing, (3) low-pressure, large diameter pump suction valves, and (4) low-pressure valves for use in discrete (multiple) pump suction lines.

The high-pressure, discrete valve concepts chosen for investigation are sleeve valves (Figures VII-C-1 and -2), poppet valves (Figures VII-C-6 and -7), rotary sleeve valves (Figure VII-C-13),butterfly valves (Figure VII-C-14), ball valves (Figure VII-C-20), and venturi valves (Figure VII-C-10) which are a special case of poppet valves. Semi-detailed drawings illustrating how sleeve, poppet, ball and venturi valves could be installed in the AJ-1 engine are shown in Figures VII-C-4, VII-C-8, VII-C-23, and VII-C-11, respectively, to give an indication of the approximate relative size of the various valve concepts in similar applications. Figure VII-C-12 shows the AJ-2 engine with venturi valves in the fuel and oxidizer discharge lines and a hot-gas TVC sleeve valve. Figure VII-C-9 illustrates the application of a 17-in. pressure-balanced angle poppet valve to the AJ-5 engine. In the case of pump discharge valves which would be integral with the pump housing, the concepts studied were the ring-gate valve, and multiple venturi valves integrated with the pump discharge diffuser (Figures VII-D-1 and -2). Figure VII-D-3 shows the application of the integrated venturi valves to the AJ-1 fuel pump. The category of large, low-pressure, single-unit pump suction valves includes the ring-gate concept (Figure VII-E-1) and the unbalanced poppet concept (Figure VII-E-2). Low-pressure

VII, A, Method of Establishing Design Criteria (cont.)

valves for multiple, discrete suction lines would be the same concepts as high-pressure valves, with the exception of structural considerations which have greater emphasis on rigidity than on stress due to pressure.

The valves categorized as discrete, high-pressure, large flow-rate valves have general physical layouts that are independent of size and pressure over a reasonable range. Analyses were performed on these valve concepts by making a simple general layout as a model for each concept and applying analytical methods to determine pressure loss, valve weight, envelope size and actuation force, all in terms of basic design parameters such as flow-rate, line size, pressure, fluid, and material properties. In this manner, formulas were derived whereby the basic design features of different valve concepts can be compared for any given set of design parameters. The relationship between envelope size as a function of line size and system pressure for the different valve concepts is presented graphically as a series of valve sketches at several line sizes and pressures for 5- 10-, and 20-in. line sizes, and system proof pressures of 1000, 3500, and 6000 psi (Figures VII-C-16, -17, and -18).

Unlike line valves, the physical configuration of integrated pump discharge valves is directly dependent on the pump design; therefore, it is difficult to design in terms of the parameters used with the line valves, such as line diameter. Preliminary design work included a scaled layout of these valves installed in the AJ-1 fuel pump housing (Figures VII-D-1 and -3). Sufficient design detail was obtained to ensure that the concepts were feasible with regard to basic physical requirements such as compatability with pump housing design, dimensions of flow passages, and mechanism of actuation. Choosing one valve concept over another can only be accomplished by a rigorous analysis of a detailed design, or by testing a prototype of the valve. As an example, the flow characteristics of a venturi type valve can be predicted analytically with reasonably accurate results.

The calculations for determining the loss through an angle poppet valve is another matter. In order to achieve a minimum loss through the poppet type valve,

VII, A, Method of Establishing Design Criteria (cont.)

a development program including experimental flow path optimization would be required. The location of the valve seat relative to the inlet port, the shape of the inlet passage, the shape of the poppet, and the position of the poppet have an effect on the efficiency of the flow. A program conducted by G. Vermes at Allis-Chalmers, Milwaukee, Wisconsin, using full-scale models and low-pressure air flow resulted in a significant improvement in the pressure drop of large steam turbine governing valves. Resistance in these optimized angle-poppet valves was reduced to a K of 0.65. Since both detailed design and experimental optimization were beyond the scope of this program, a compromise approach was selected. This consisted of relying on past experience and available development test results on valves subject to similar flow conditions. Problem areas considered include distortion of non-symmetrical sections due to pressure forces and temperature gradients, the effect of high-frequency acceleration loads on moving parts (i.e., vibration), thermal stresses during start and shutdown transients, sensitivity to the presence of ice or foreign particles, binding of sliding parts due to small distortions or nonuniform friction drag, and other similar problems.

The approach taken to analyze different valve concepts for large, low pressure applications was to concentrate on the two primary factors considered in selecting a valve design for pump suction control (excluding seal considerations). These factors are (1) valve length as it directly affects overall vehicle length, and (2) valve pressure loss which directly affects tank pressurization and tank weight. Because these valves are for relatively low pressures, weight considerations are of little consequence relative to the previously mentioned factors. The general valve concepts considered most feasible for pump suction application are: ring-gate valves, ball and butterfly valves, and unbalanced poppet valves. Seal considerations are of primary importance in selecting a valve when the application requires a positive shut-off; but to accurately describe the conditions the seals must tolerate would require a very detailed design and analysis of deflection. This type of study is not within the scope of this program; therefore, only the relative magnitude of the sealing problem as it is related to the basic design concept will be discussed qualitatively.

VII, Valves (cont.)

B. VALVE CONCEPT SELECTION

The following information is presented as the basis for selecting the best valve for a particular set of engine requirements. It must be emphasized that the valve choice for a particular application depends to a great extent on the current technology in such areas as seal design, seal material, fabrication techniques, available materials, etc. It would be unwise to recommend a particular valve concept on the basis of limitations imposed on other concepts by current technology, without first determining if it is possible to overcome those limitations.

In many instances, the choice of a valve configuration evolves from the limitations imposed by the seal functions, with primary concern placed on the main shutoff seal or seals that control leakage between the inlet and outlet ports of the valve. Development work on seals in recent years has been directed toward the development of seal materials that will (1) withstand the corrosive effect of modern propellants and (2) maintain functional integrity (primarily sealing) at cryogenic propellant temperatures and at the high temperatures encountered with hot-gas systems. In applications where the temperature is in the range of -340 to +450°F, elastromeric compounds (i.e., Teflon, Kel-F, Viton, etc.) are commonly used for both static and dynamic seal functions. Seals constructed of these materials are available in a variety of mechanical configurations, but to date they all share the common limitation of requiring controlled clearances between mating parts. The allowable clearances are determined through development tests in which the seal material and configuration are optimized. For example, a 5-in. steel cylindrical part that is pressurized and stressed to 50,000 psi deflects about 0.008 in. diametrically; this, added to a 0.002-in. initial clearance, results in a total radial clearance of 0.005 in. This is about the maximum gap that present-day configurations of nonmetallic seals can tolerate before extrusion or other permanent deformation occurs at the high pressures considered in this study. A wall thickness four times that of the 5-in. dia cylinder and resulting in a 50% reduction in design stress level is required to maintain the same deflection in a 10-in. dia cylinder. It is also

VII, B, Valve Concept Selection (cont.)

costly to hold tolerances on initial radial clearances as small as 0.002 in. in larger valve parts. Consequently, as the size of the valve increases, the design stress level of moving, pressure-loaded parts must be optimized in conjunction with the wall thickness to reduce deflection and the stringency of tolerances of the parts. Alternative design techniques that utilize insert rings or sleeves pressure-balanced to reduce hoop strain differentials between sealed sliding surfaces must be developed. Modern high-strength materials allow the designer more stress tolerance to work with; however, cycle life or valve endurance would ultimately control final design criteria. The present seal clearance requirements also impose prohibitive tolerance restrictions on the initial clearance of large mating parts in an unloaded state. For example, a 0.005-in. clearance having a tolerance of +0.005 in., -0.000 in. (for 0.010 in. maximum clearance) is a tolerance of 0.1% on a 5-in.-dia part; but for a 30-in.-dia part (e.g., a sleeve of a ring-gate valve) this same tolerance is only 0.017% of the diameter. In applications where temperature gradients exist, the deflection in parts having different temperatures must also be considered. A steel valve with concentric cylindrical parts of 5 in. dia will undergo changes in the diametral clearance of about 0.003 in. for each 100°F temperature differential between parts. A 30-in. valve (e.g., the AJ-1 ring-gate fuel-pump discharge valve) would have a corresponding diametral deflection of 0.018 in. for the same temperature differential, which is well beyond the clearance gap that present-day nonmetallic seals can tolerate at high pressure.

Advanced technology efforts associated with sealing problems are presently the subject of a wide variety of programs. Several current contracts under which studies applicable to rocket engine valve sealing problems are being conducted are listed in Appendix A. Contract AF 04(611)-8020, "Analytical Techniques for Design of Static, Sliding, and Rotating Seals for Use in Propulsion Subsystems," is the only program that seriously treats sliding seals such as those discussed in the preceding paragraph. P. Bauer of the Illinois Institute of Technology Research Institute, the agency conducting this program for Edwards Air Force Base, has indicated that a significant breakthrough in the sealing properties of elastomers, plastics, or metals

VII, B, Valve Concept Selection (cont.)

will be required in order to meet the low leakage restrictions of high-pressure sliding seals for applications with propellants. Investigations by the institute to date have indicated that contact stresses of 1000 psi are required to significantly alter the leakage rates, but that sliding seals yield little improvement in leakage rates at stresses above 1000 psi.

Since current seal technology severely restricts the applicability of several attractive valve concepts, it was assumed in the following discussion of valve concepts that future development work in seal technology will produce techniques for sealing high pressures in large valves and for providing compensation for differential-expansion clearances. The results of the valve concept studies in relation to seal requirements will be presented only to illustrate the differences in general seal requirements for each valve concept, (e.g., number of dynamic seals, size of seals, effect of pressure deflection on sealed parts, etc.).

Analyses of the valve concepts used information from development testing reports and consultations with experienced valve designers and manufacturers. It should be noted that an individual's previous experience with various valve types frequently influences the selection of a particular valve type more than any specific technical consideration. Several experienced design engineers, any of whom might be in a position to prescribe the valve concept for a rocket engine application, were asked what valve type they would recommend for valves up to 20 inches in diameter handling pressures up to 600 psi. The replies included recommendations for poppet, venturi poppet, ball, and sleeve (providing adequate seals were available) types. Only butterfly valves were totally excluded.

In the following discussion of various valve concepts the same presentation order is followed to facilitate comparisons; i.e., a - description, b - envelop, c - sealing, etc.

A. Fluid-Transfer Considerations

VII, C, High-Pressure Line Valves (cont.)

2. Poppet Valve Balanced and Unbalanced

a. The poppet type valve can be designed either in a pressure-balanced (Figure VII-C-6) or unbalanced configuration (Figure VII-C-7). Figure VII-C-8 shows how poppet valves might be applied to the AJ-1 engine. The choice relies mainly on a trade-off between size considerations of main seals and actuators. The unbalanced poppet design has only one main seal between the inlet and discharge ports, which is the poppet-to-seat seal in the shutoff position. The penalties incurred in this choice of configuration are (1) the requirement for an actuator capable of supplying the necessary large force to oppose the pressure force against the poppet seat area (for a 5-in. 3000-psi valve this force is 60,000 lb), (2) high unit seat loading, and (3) friction and inertial forces. The balanced poppet design, on the other hand, requires a large dynamic seal in addition to the main poppet-to-seat seal. The balanced poppet design is therefore very similar to the sleeve valve design with the exception that one of the seals (poppet-to-seat) is not a sliding seal. An application of a 17-in. pressure-balanced angle poppet valve to the AJ-5 engine is shown in Figure VII-C-9. Some poppet valves replace the sliding seal with a welded bellows, which provides a hermetic seal. Such designs would be highly desirable where adequate high-pressure welded bellows are available. The actuator force requirements of the balanced poppet valve are similar to those for the sleeve valve concept.

b. The envelope size for the in-line, pressure-balanced poppet valve is relatively small compared to the balanced valve, because the actuator can be integrally designed within the valve body. The unbalanced poppet valve is usually designed as an angular unit to facilitate placement of the large actuator in line with the axis of the poppet motion. The small poppet actuator shaft of the unbalanced poppet valve also facilitates the flow pattern normally associated with angle valves and eliminates the large external collection torus that is generally required for an angle sleeve valve. The compactness of the angle valve will in most cases be offset by the large actuator that is externally mounted aft of the poppet.

VII, C, High-Pressure Line Valves (cont.)

c. The seal requirements previously discussed are usually the predominant factor to be considered when choosing the basic valve configuration. The unbalanced poppet valve requires only one main poppet seat seal. One smaller sliding shaft seal is required if the valve is externally actuated. Design simplicity and generous manufacturing tolerances of the self-centering, poppet-type seal make it attractive for large-diameter, high-pressure applications, even though the actuator requirements are considerably less favorable. The balanced poppet concept requires an additional dynamic sliding seal between the poppet and body, thus restricting the design to small clearance and deflection limitations such as with the sleeve valve.

d. The pressure loss of the in-line poppet-type valve can be made quite small if the stroke length is sufficient to allow gradual rather than abrupt turning of the flow. This is especially important when the poppet seat is at the discharge side of the valve. An in-line poppet valve can be designed to have a calculated loss coefficient as low as 0.4 (Table VII-C-1). Angle poppet valves have been built in flight-weight configurations having a loss coefficient of about 1.2. (For steam turbine valves, coefficients of 0.65 to 1.0 have been achieved with careful design.)

e. Poppet-type valves are excellent for flow modulation because the characteristic curve can be varied by simply shaping the contour of the poppet plug, with a corresponding adjustment of stroke to achieve the maximum flow area (Figure VII-C-5). Unbalanced poppet designs have a disadvantage in that the actuator force varies with supply and load pressures; but this problem can be minimized by employing an actuator of sufficient power. For good response characteristics, the poppet plug may be shaped so that the perimeter regulates the flow area; in such a design, the stroke for full flow is only one-fourth of the seat diameter.

VII, C, High-Pressure Line Valves (cont.)

f. In-line poppet valves, like in-line sleeve valves have symmetrically shaped (generally cylindrical) parts that are desirable from a standpoint of thermal stress and fabrication. This is of particular significance in cryogenic or hot-gas applications where thermal distortion may exceed pressure load deflections.

3. Venturi Valve

a. The purpose of applying venturi-type inlet and discharge sections to a particular valve is to reduce the size of the critical valve parts, such as the poppet and seat in a poppet type valve by using the venturi throat as the valve seat (Figure VII-C-10). Since the cross-sectional area at the throat of a venturi is considerably less than the line section area, the static pressure force on unbalanced valve elements will be correspondingly smaller (by the ratio of line to throat diameter squared). Venturi valves have found numerous applications on missiles, spacecraft, and rocket engine test facilities, but no known applications have been made to the control of large flow rates within actual engine systems. Figures VII-C-11 and -12 shows how discrete venturi poppet valves could be applied to the AJ-1 and AJ-2 engines.

b. The envelope of venturi-type valves is characterized by small diameter (in some cases no larger than line size) but relatively long length. The length is a result of the divergent diffuser section, which must have an included angle of divergence of no more than 8 to 10° from the venturi throat to the line size at the valve outlet for efficient conversion of kinetic to static pressure. The relationship between valve length, throat-to-line diameter ratio and pressure loss is given in Table VII-C-1 and Appendix D.

c. The seal requirements of venturi valves are dependent on the configuration of the valve closure element used in the venturi throat. The use of an unbalanced, single seal (poppet-to-seat) poppet has been successful in small applications of this valve principle (up to about 3-in. line size); there are no obvious size limitations. The design simplicity coupled with the symmetrical configuration make this type of valve particularly attractive for use where high pressures and wide temperature ranges are expected.

VII, C, High-Pressure Line Valves (cont.)

d. Pressure losses with venturi valves have been analyzed for several values of throat-to-line diameter ratios. Loss coefficients are shown in Appendix B. The loss will never be less than 8% of the kinetic head at the throat; this is the minimum loss to be expected of a well-designed divergent line section. Actual total loss will be approximately 12% of the kinetic energy at the throat because there is a 2 to 4% entrance loss caused by gradual convergence. Under conditions of low throat-to-line diameter ratios, the throat static pressure is practically zero in comparison to the total pressure in the line when the venturi is cavitating, as is sometimes desired in order to provide flow control.

e. By designing the venturi so it cavitates, the valve can be used to limit the flow rate in proportion to upstream pressure regardless of fluctuations in downstream pressure over the range of 18 to 88% of upstream pressure. A shaped pintle can be used in a cavitating venturi valve to vary the flow rate by varying the throat area. When the venturi is cavitating, the overall pressure loss will be about 12% of the total upstream pressure.

4. Rotary Sleeve Valve

a. The rotary sleeve valve is a pressure-balanced type unit that functions through rotation of the sleeve rather than by translatory motion of an element. No large flow rate applications of this type valve in liquid rocket engines were found, but the concept is presently being studied for certain solid rocket thrust vector control applications.

b. The envelope of the rotary sleeve valve is quite short in comparison with most other valves. The outside diameter of the valve housing corresponds to that of the sleeve and in-line poppet type valves.

c. The sealing problem of the rotary sleeve has virtually prohibited its use in applications where leak-tight closure is required. The distortion of the sleeve because of pressure forces is considerable where large port areas are necessary; thermal gradients across such a rotating seal area would make axial and circumferential sealing of the cylindrical surface extremely difficult. The lack of this type valve in large, high-pressure applications is evidence.

d. Pressure losses in this configuration are greater than with most other valve types because of the layout of the sleeve ports that is necessary to allow closure. The circumferential length of the ports must be designed to permit sufficient material width between the ports to effect closure when the sleeve is rotated (Figure VII-C-13). This dictates that the maximum full-open width must total less than one-half of the circumference.

e. The rotary sleeve valve has the desirable characteristic of a linear relationship between port area and sleeve rotation (if rectangular ports are employed). If desired, variation in flow characteristics can be obtained by shaping of the port. The response of this type of valve is quite rapid, considering that only a small rotation of the sleeve is required from the closed position to effect full-open operation. Sleeve rotation of 45° is necessary for a four-port valve. Increasing the number of ports would further reduce the rotation required, but more circumferential port length is then lost to accommodate the greater number of seals.

f. The most attractive feature of the rotary sleeve valve is that the surfaces and seals that move in the valve are not exposed to the flow where the valve is open. Because of its attractive features with regard to control, a seal development program should be considered for this type of valve if present sliding seal studies indicate such seals are feasible.

5. <u>Butterfly Valve</u>

a. The butterfly valve (Figure VII-C-14) is not a pressure-balanced valve. When the valve is closed, the entire static pressure force on the blade must

VII, C, High-Pressure Line Valves (cont.)

be carried by the shaft and bearings. This fact alone limits the usefulness of this concept for large high-pressure applications. The limiting factor is the deflection of the blade, shaft, and bearing assembly caused by the pressure load in the closed position. Ultrahigh strength materials could be utilized for the shaft and blade to prevent their failure, but the elastic moduli of these materials, being the same as conventional metals, would still allow seal and bearing failure from deflection. The use of this concept in the size and pressure range considered would require considerable advances in seal and bearing technology in order to tolerate large deflections. Increase of shaft size to reduce deflection results in an increased blade section that in turn requires local cavity enlargement to maintain flow area. This in turn increases the exposed blade area and loading. Butterfly valves have found application in many state-of-the-art rocket engine systems such as Atlas and Titan. Figure VII-C-15 shows some of the butterfly valves developed at Aerojet-General for the Titan I and Titan II engines.

b. The very short envelope length (Table VII-C-1; Figures VII-C-16, -17, and -18) required by the butterfly valve is one of its most attractive features. The necessary bearings and large external actuator require considerably more space than the valve, which requires approximately one line diameter of in-line length.

c. The main blade-to-housing seal, the shaft seal, and the bearings are the primary factors limiting the usefulness of this valve concept. Failure of these components is largely the result of deflections of the butterfly blade and shaft.

d. The pressure losses from butterfly-type valves is dependent on the design pressure. A high-pressure valve requires increasing of the blade diameter to regain flow passage area and results in an indirect flow path through the valve. These factors, in addition to deflection restrictions, must be regarded when considering this type of valve for large high-pressure applications.

VII, C, High-Pressure Line Valves (cont.)

e. The characteristic flow curve of butterfly valves is an equal-percentage curve for the first 50% of rotation, and nearly linear for the remainder (Figure VII-C-19). Although this curve may be desirable in some applications, it cannot be significantly altered by shaping of the blade as can the flow characteristic curve of sleeve and poppet valves. There is a considerable as well as variable torque applied to the blade by the flow stream when the valve is opening or closing that inhibits accurate positioning of the blade for flow modulation (Appendix F).

f. A number of aspects of the butterfly valve make it undesirable for large high-pressure applications. The irregular stress pattern in the body where the shaft bearings are located and the unsymmetrical bearing loads on the housing require large thick sections in the valve body. The complex distortion pattern of the blade and shaft results in complex sealing and bearing requirements. The problems associated with butterfly valves outweigh the desirable features when compared to other valve concepts being considered for high-pressure engine designs.

6. Ball Valve

a,b. Ball valves (Figure VII-C-20) like butterfly valves are not pressure balanced. The total force from system pressure on the enclosed seal area must be supported by shafts and bearings located in the valve body. The stress and deflection resulting from load distribution results in ball, shaft, bearing, and housing dimensions that are large in comparison with other valve concepts. This may be of little consequence in ground systems, but it is of considerable importance in flight-engine applications.

Ball valves have been used in numerous small high-pressure applications. In addition, AiResearch Manufacturing Division of the Garrett Corporation has produced 17-in. suction valves (185 psi proof) for the Saturn vehicle and 8-in. ball valves for thrust vector control on the 120-in. United Technology Center solid rocket motor for the Titan III rocket vehicle. This latter

VII, C, High-Pressure Line Valves (cont.)

valve has a membrane providing for zero leakage of N_2O_4 at 1000 psi for up to 45 days; secondary seals are used to limit leakage to 200 cc/min at a pressure of 900 psi after the membrane is ruptured by initial opening of the valve.

c. Ball valves require one main dynamic seal between the spherical ball surface and the body and a smaller shaft seal on the actuation shaft. In large high-pressure applications, the main seal is pressed into contact with the ball by a spring and pressure-loaded sleeve (which must also be sealed). In this case, the deflection limitations would be imposed by shaft seal tolerances, by bearing failure, and by distortion of the spherical surface with which the main seal is in contact. It is possible but not practical to control deflections by designing the ball, shaft, and housing for low stress levels. This is illustrated by the comparative illustrations shown in Figures VII-C-16, -17, and -18.

d. The pressure loss of ball valves is the lowest of any valve type; with proper design it can be made practically zero (equivalent to an equal length of straight pipe). The low-pressure loss in some instances will make the large valve and actuator envelope an acceptable penalty for obtaining the minimum pressure drop characteristic.

e. The flow characteristic curve of the ball valve (Figure VII-C-21) cannot be feasibly changed without degrading the excellent pressure drop features. The valve requires a 90° rotation to go from full-open to full-closed operation. The seal and bearing friction torque (Appendix F and Figure VII-C-22) is considerable for large high-pressure valves.

f. The general design of the ball valve is a simple, rugged, and reliable concept that features a very minimum of pressure loss; however, the unbalanced load imposed by the system pressure requires large and heavy valves for large high-pressure applications.

TABLE VII-C-1

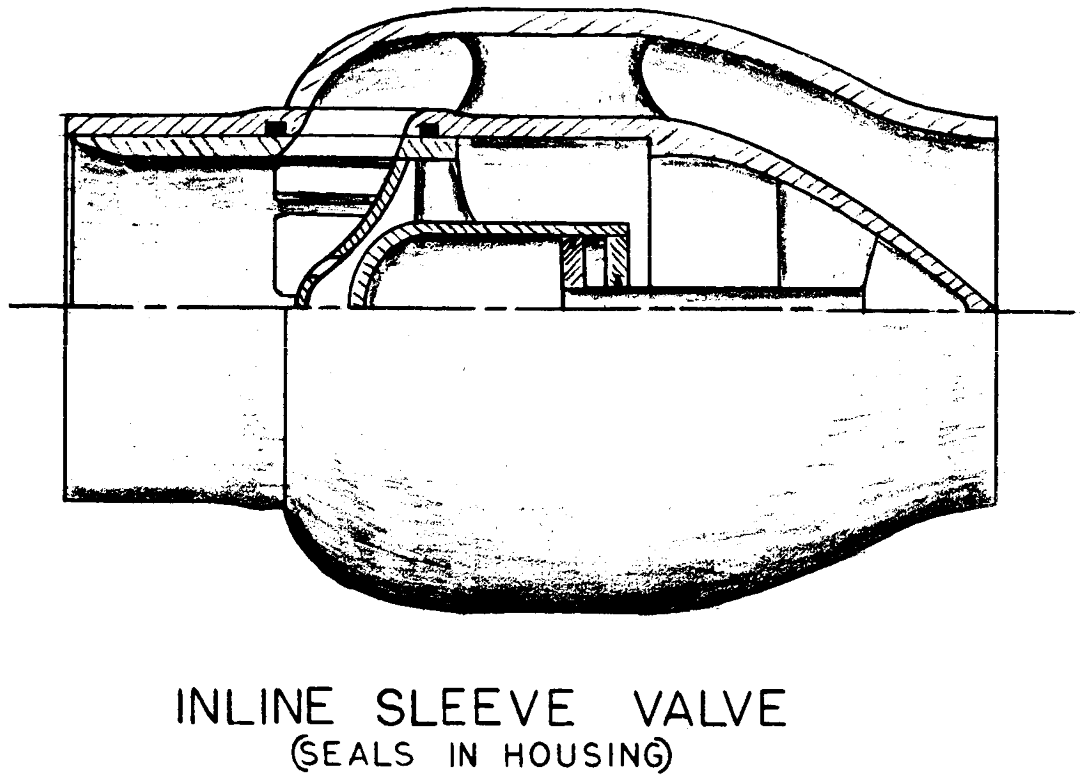
RESULTS OF PARAMETRIC STUDIES OF DISCRETE, HIGH PRESSURE VALVES

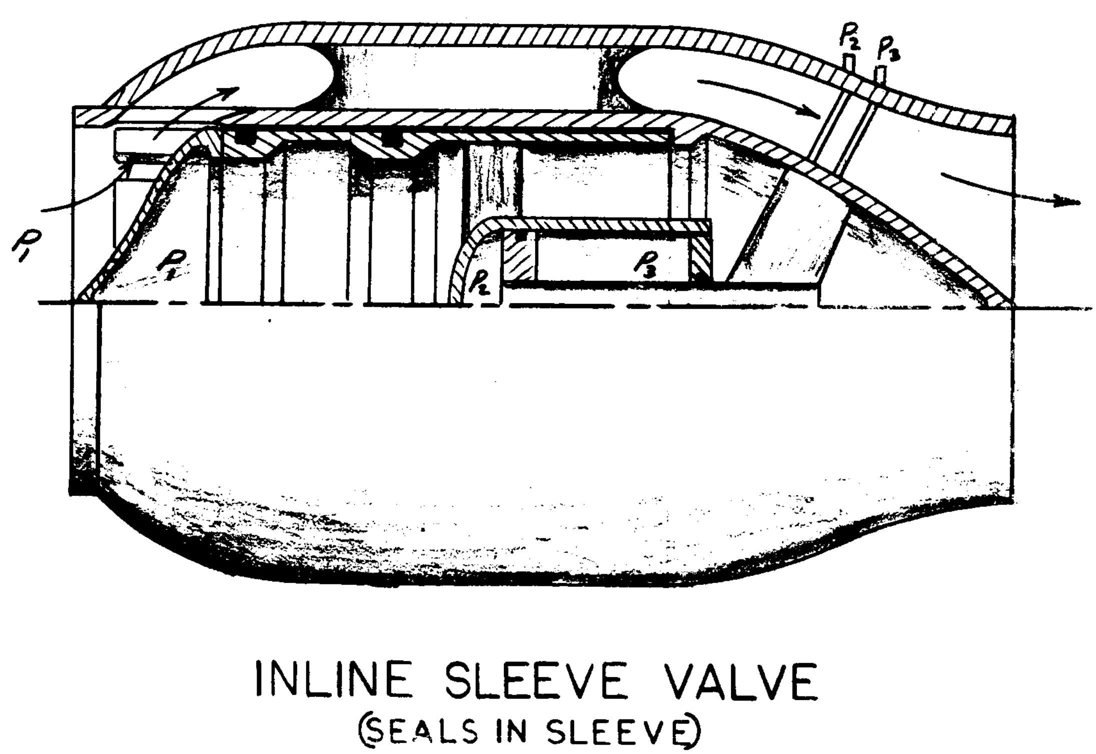
Valve Type	Figure	Valve Weight Without Actuator	Appendix	Valve Pressure Loss	Appendix	Minimum Valve Envelope Dimensions' without External Actuator (Appendix G)
Inline Sleeve Valve	VII-C-1	$15.62 \frac{PD^3 \rho'}{S}$	B, I	$1.83 \frac{\dot{w}^2}{\rho D^4}$	A, II	Cylindrical L = 3D, OD = 1.75D
Angle Sleeve Valve	VII-C-2	$18 \frac{PD^3 \rho'}{S}$		$4.24 \frac{\dot{w}^2}{\rho D^4}$	A, III	Cylindrical L = 2D, OD = 2.5D
Inline Poppet Valve (Pressure balanced)	VII-C-6	Similar to sleeve valve		Similar to sleeve valve		Cylindrical L = 3D, OD = 1.75D
Angle Poppet Valve (Not pressure-balanced)	VII-C-7			Approx $4.4 \frac{\dot{w}^2}{\rho D^4}$	*	Depends on actuator used Typical: Cylindrical L = 4D, OD = 2.2D (From Report AFFTC-TR-60)
Venturi Inlet and Discharge Sections	none	$\frac{9\, PD^3 \rho'}{S}(1-R^2)$	C	$\left[.02 + .08\,(1-R^4) + K_v\right] \frac{3.64\, \dot{w}^2}{\rho D^4}$	C	Depends on the valve used
Rotary Sleeve Valve	VII-C-13	$10.46 \frac{PD^3 \rho'}{S}$	B, II	$4.96 \frac{\dot{w}^2}{\rho D^4}$	A, X	Cylindrical L = 2D, OD = 1.75D
Butterfly Valve	VII-C-14	$\left[6.9 \times 10^{-7} P^{3/2} + \left(\frac{3900}{S} + .0044\, P\right) + 1.05 P^{3/2}/S\ (1.25\, P/S)\right] \times \rho' D^3$	B, VI	$\left[\frac{2.24}{.785 - .006P - \frac{4400}{S}} - 3.63\right] \frac{\dot{w}^2}{\rho D^4}$	A, VIII	Cylindrical L = 1.1D, OD = 1.5D
Ball Valve	VII-C-20	$1.84\, \rho' D^3 \left(1 + \frac{7P}{S}\right)$	B, VII	$.364 \frac{\dot{w}^2}{\rho D^4}$	A, IX	Cubic L = 2D $.66 \times 10^{-3}$ P + 1.6 on a side
Venturi Valve (Poppet)	VII-C-10	$\left[4.56\,(1-R^2) + 6.8R + 16.6R^3\right] \times \frac{\rho' PD^3}{S}$	B, III	R / ΔP: .8 — $1.89\, \dot{w}^2/\rho D^4$ .6 — 3.92 " .4 — 17.6 "	A, VII	Cylindrical L = 3D (1.9 - .9R), OD = 1.1D to 1.5D

*Approx pressure loss determined by flow test of actual valves and scale models.

Nomenclature

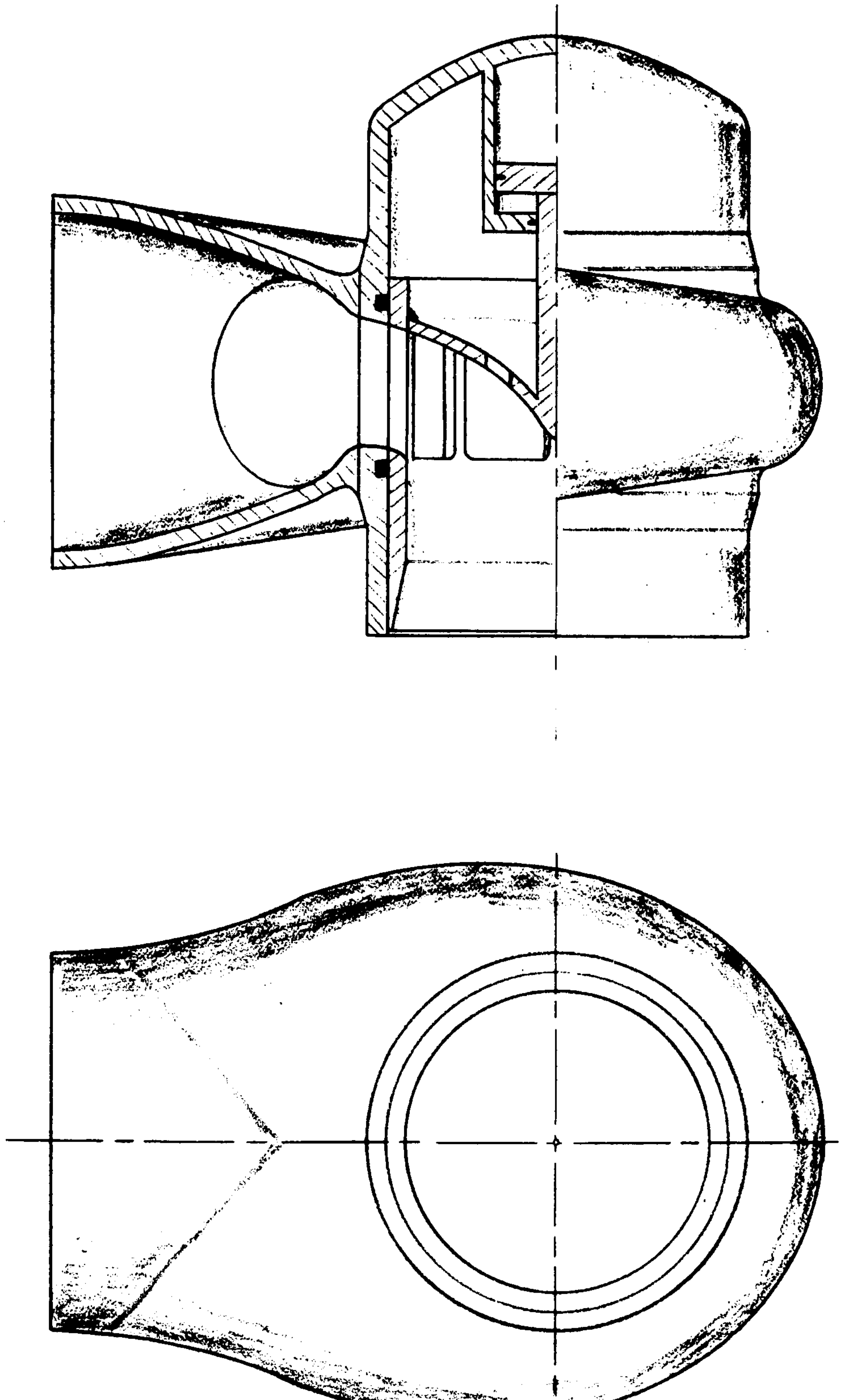
D = valve line size--inside dia, in.
L = envelope length, in.
OD = envelope outside diameter, in.
P = proof pressure, psi
R = ratio of throat-to-line inside dia (venturi valve)
S = stress in valve parts at proof pressure, P, psi
$\dot{w}$ = flowrate, lb/sec
ρ = fluid density, lb/ft^3
ρ' = material density lb/in.^{3}





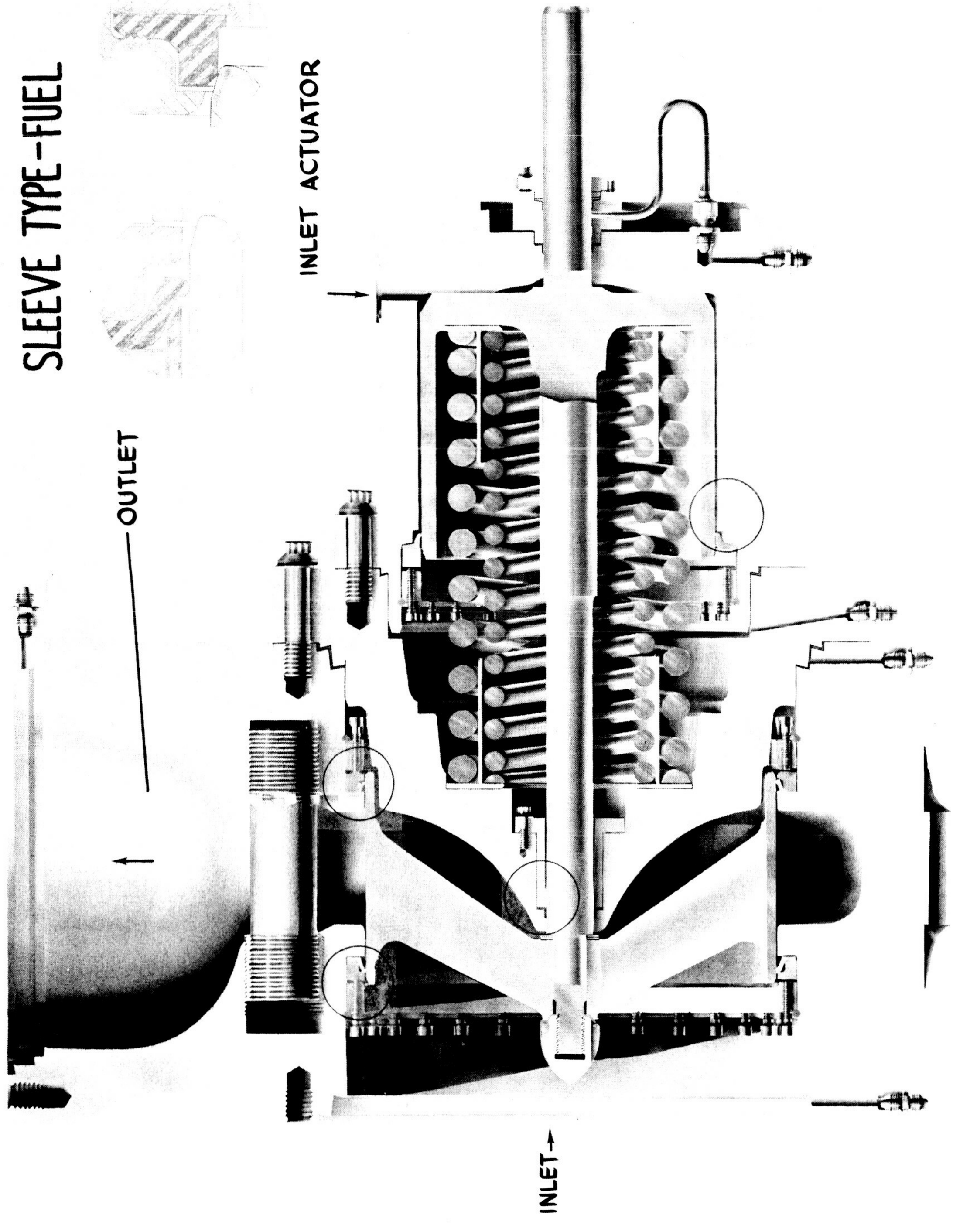
Inline Sleeve Valve

Figure VII-C-1



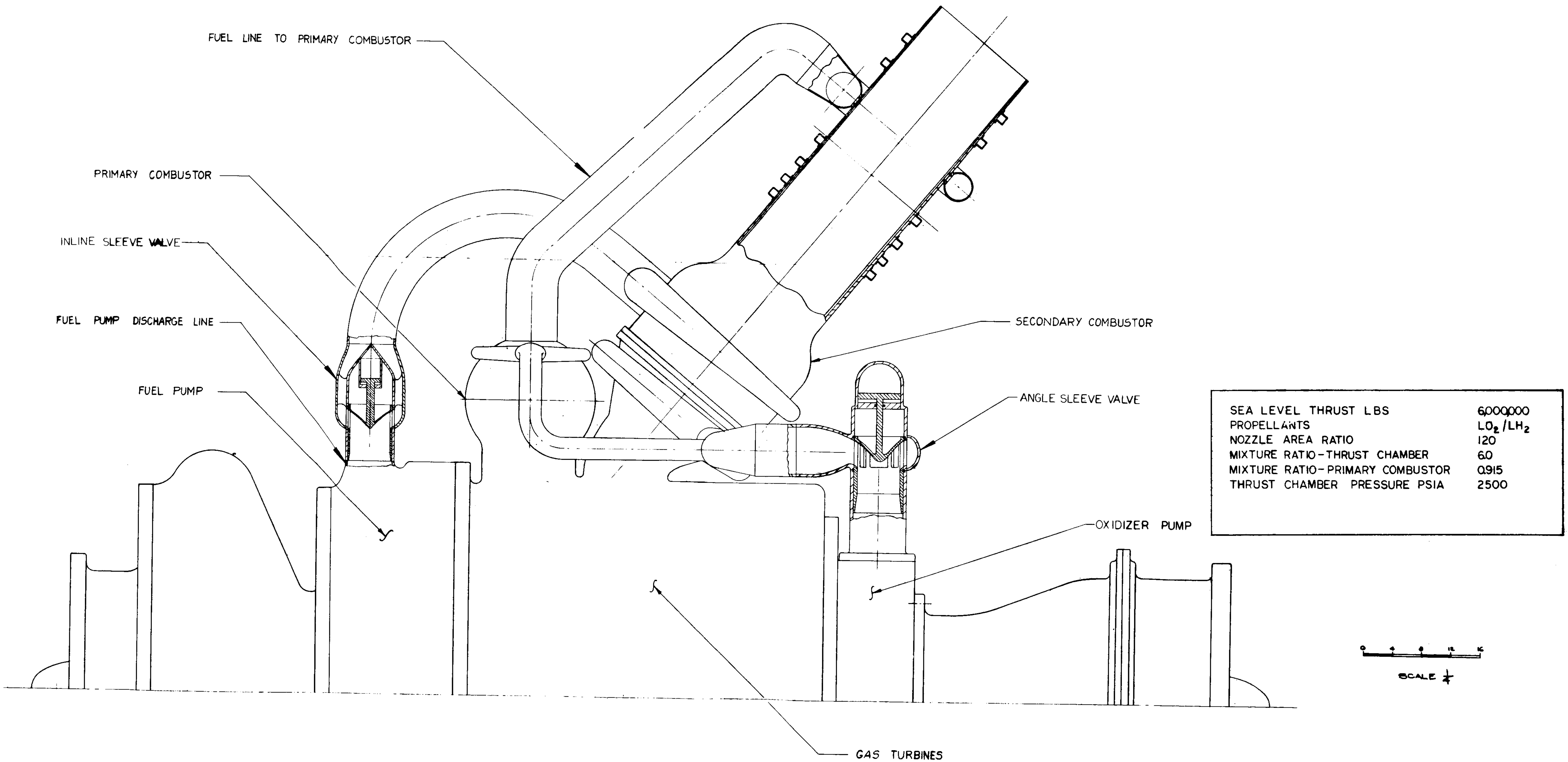
Angle Sleeve Valve

Figure VII-C-2



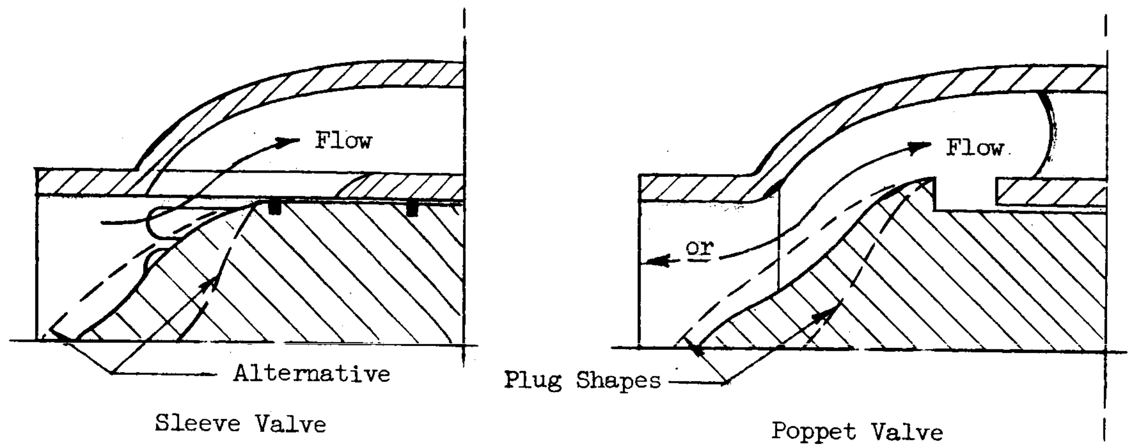
M-1 Thrust Chamber Fuel Valve

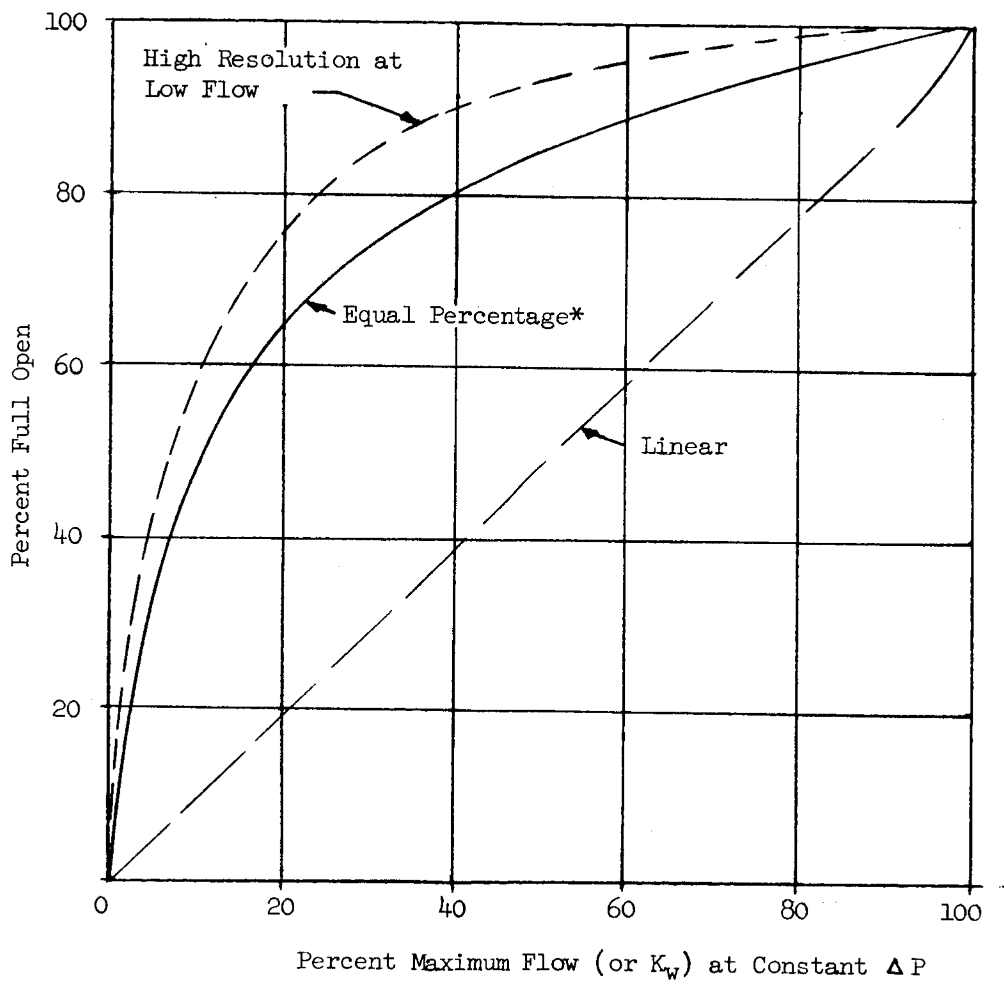
Figure VII-C-3



AJ-1 Engine with Sleeve Valves

Figure VII-C-4

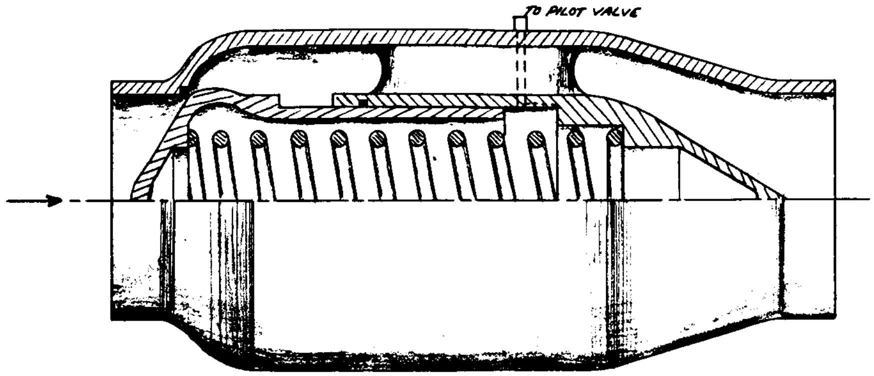




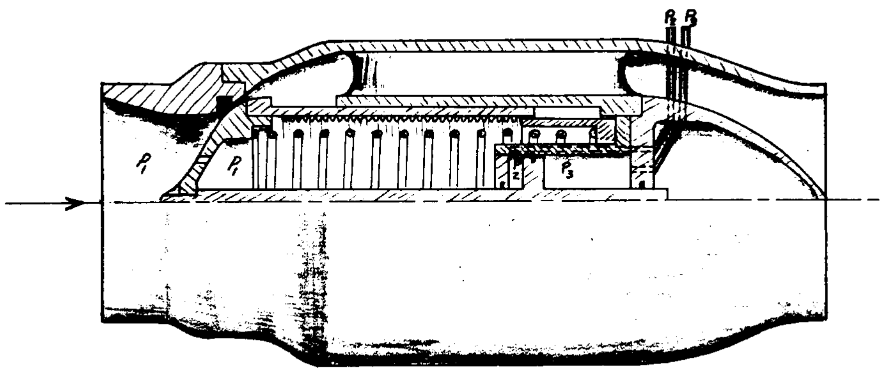
* Equal increments of stroke produce equal percent change in flow rate.

Sleeve and Poppet Valve Flow Modulating Characteristics

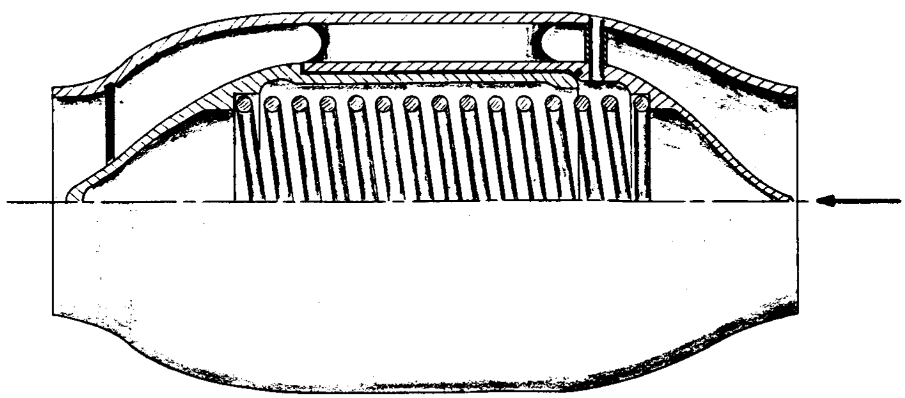
Figure VII-C-5



INLINE POPPET VALVE
(NON - MODULATING)



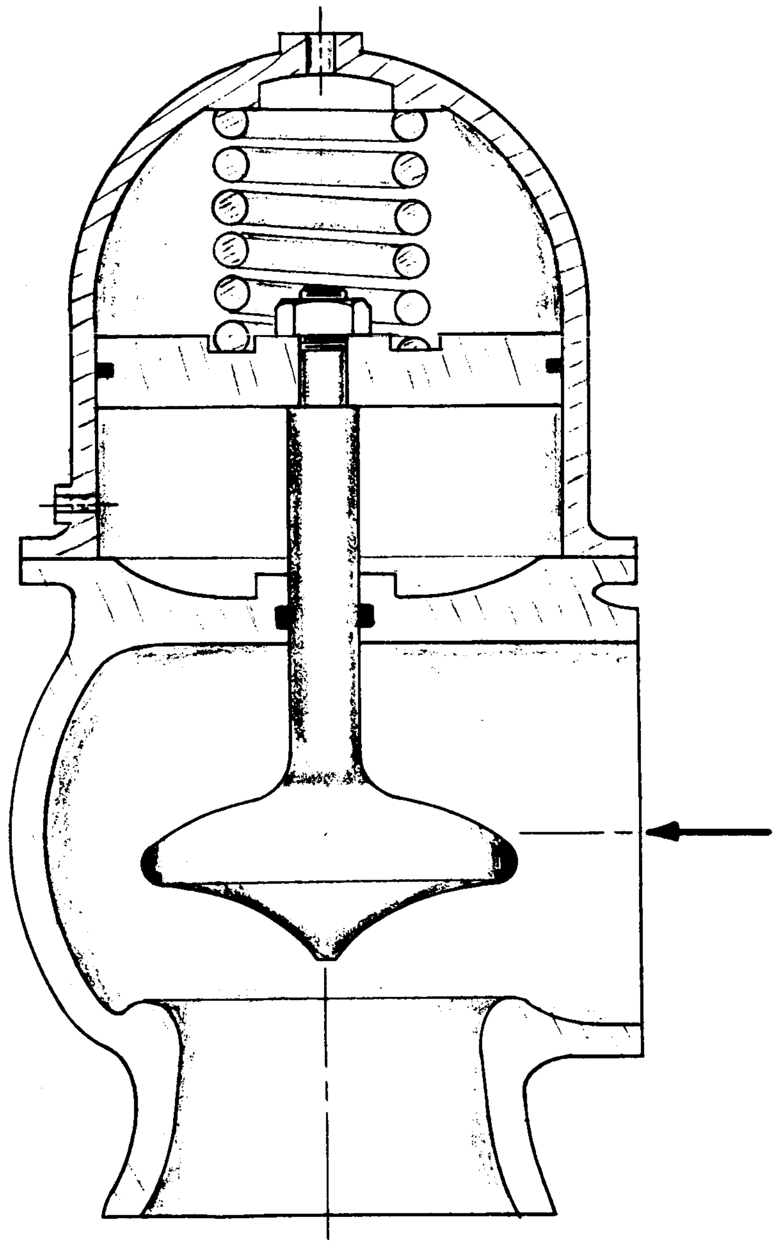
INLINE POPPET VALVE
(MODULATING WITH BELLOWS SECONDARY SEAL)



INLINE POPPET VALVE
(NOT PRESSURE BALANCED)

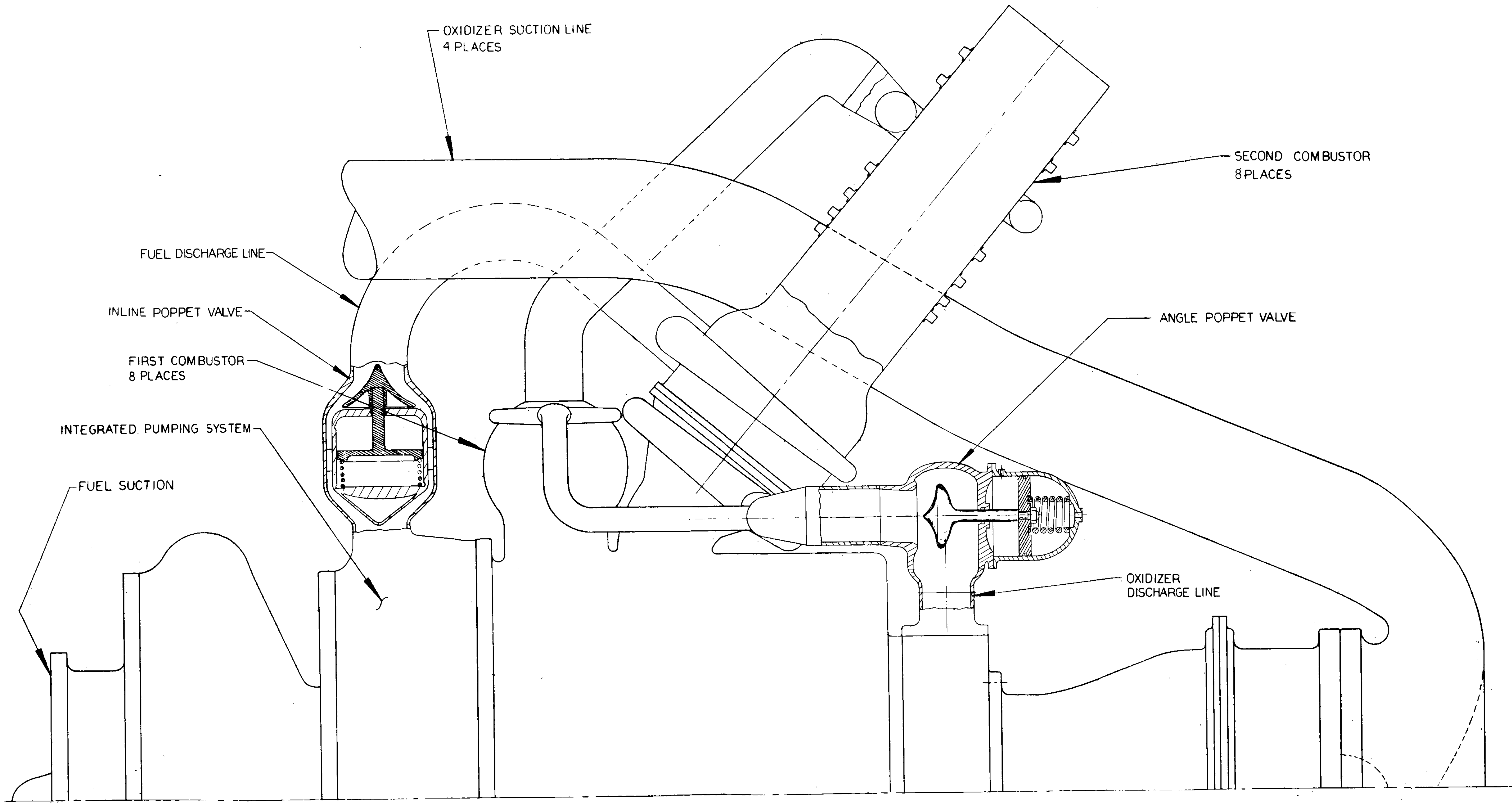
Inline Poppet Valves

Figure VII-C-6



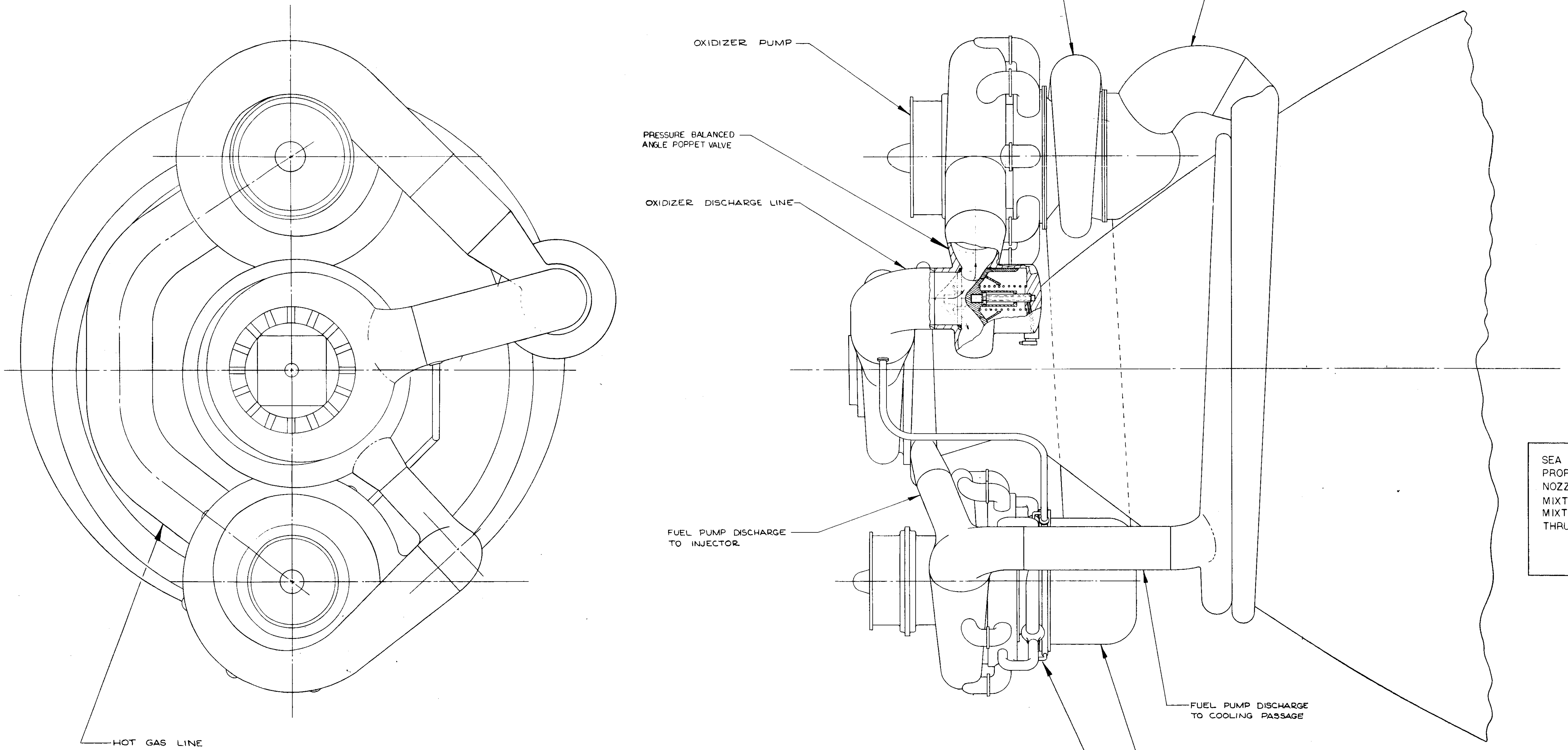
Angle Poppet Valve

Figure VII-C-7



AJ-1 Engine with Poppet Valves

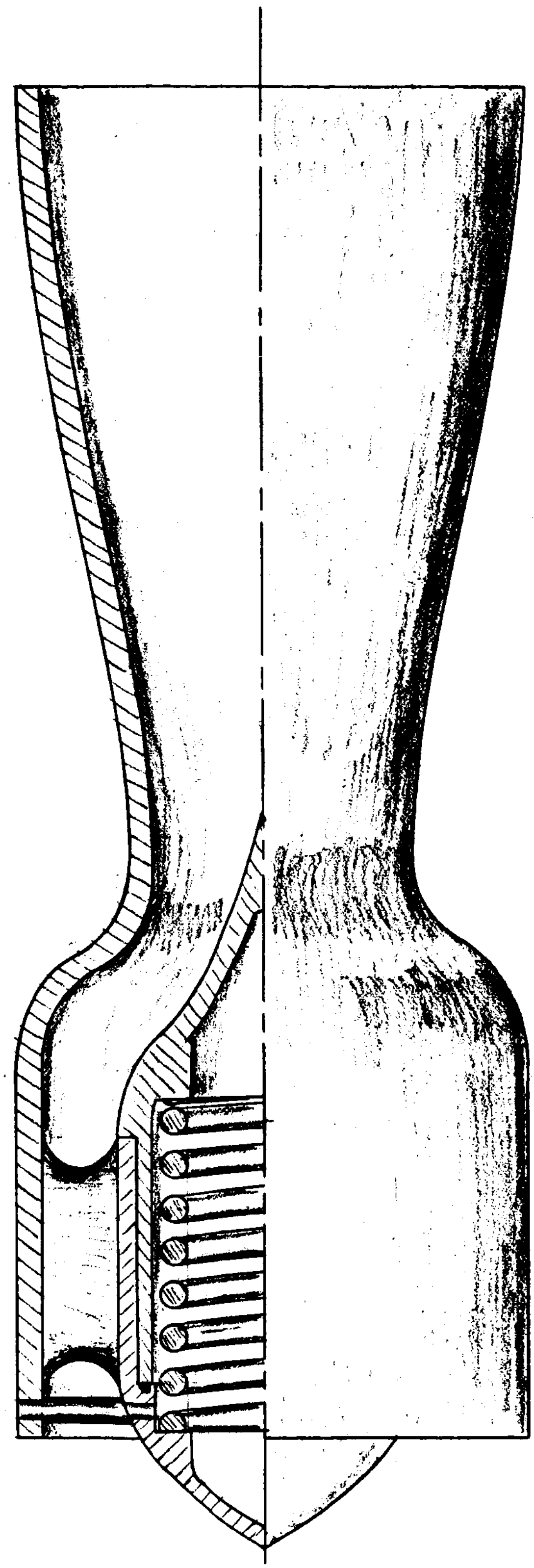
Figure VII-C-8



SEA LEVEL THRUST LBS	6,000,000
PROPELLANTS	LOX / LH_2
NOZZLE AREA RATIO	50
MIXTURE RATIO-THRUST CHAMBER	6.0
MIXTURE RATIO-GAS GENERATOR	0.915
THRUST CHAMBER PRESSURE PSIA	3400

AJ-5 Engine with Balanced Poppet Valves

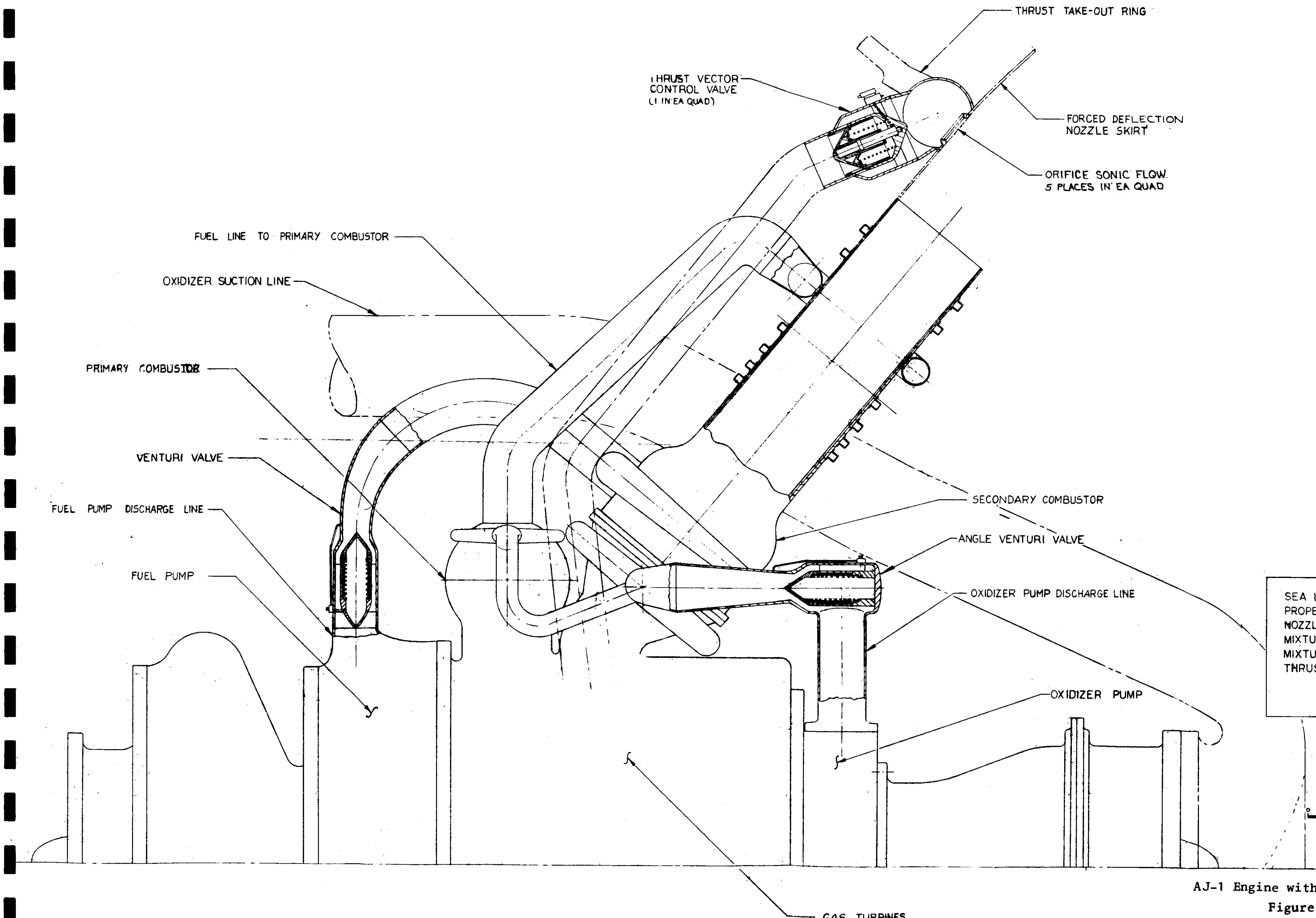
Figure VII-C-9



NON-MODULATING

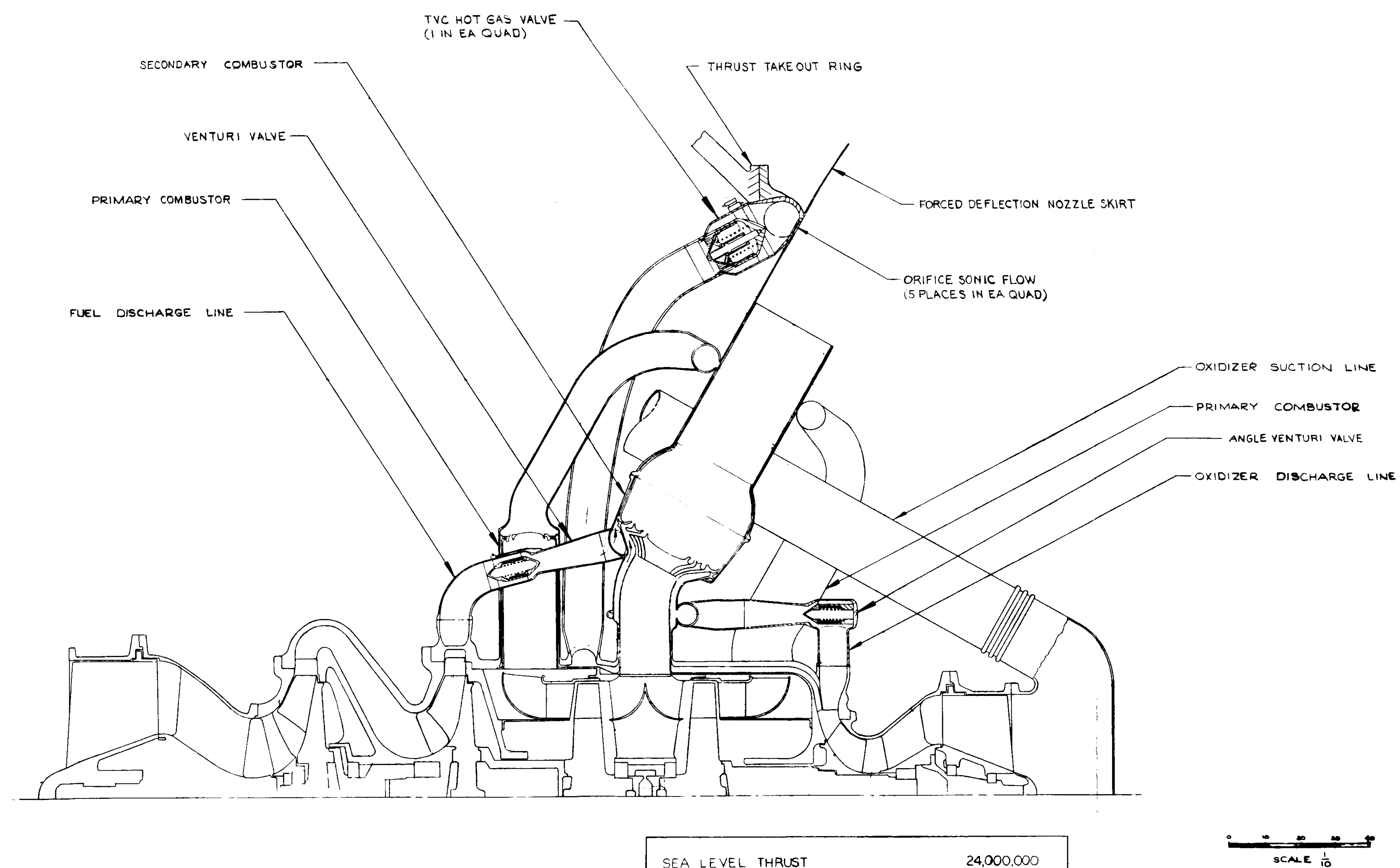
Venturi Valve

Figure VII-C-10



SEA LEVEL THRUST LBS	6,000,000
PROPELLANTS	LO_2/LH_2
NOZZLE AREA RATIO	120
MIXTURE RATIO-THRUST CHAMBER	6.0
MIXTURE RATIO-PRIMARY COMBUSTOR	0.915
THRUST CHAMBER PRESSURE PSIA	2500

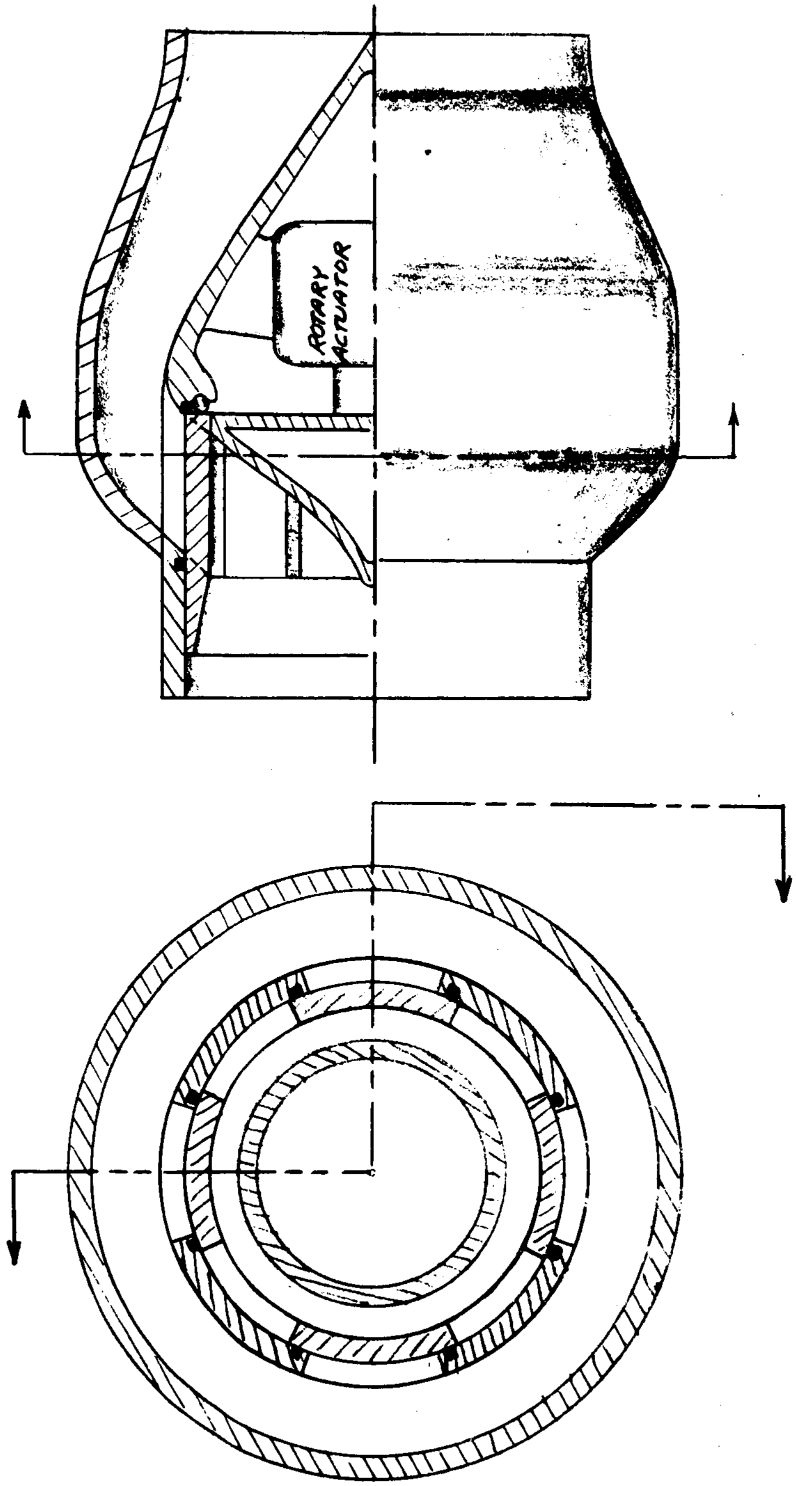
AJ-1 Engine with Venturi Valves
Figure VII-C-11



SEA LEVEL THRUST	24,000,000
PROPELLANTS	LO_2/LH_2
NOZZLE AREA RATIO	120
MIXTURE RATIO THRUST CHAMBER	6.0
MIXTURE RATIO PRIMARY COMBUSTOR	0.915
THRUST CHAMBER PRESSURE, PSIA	2500

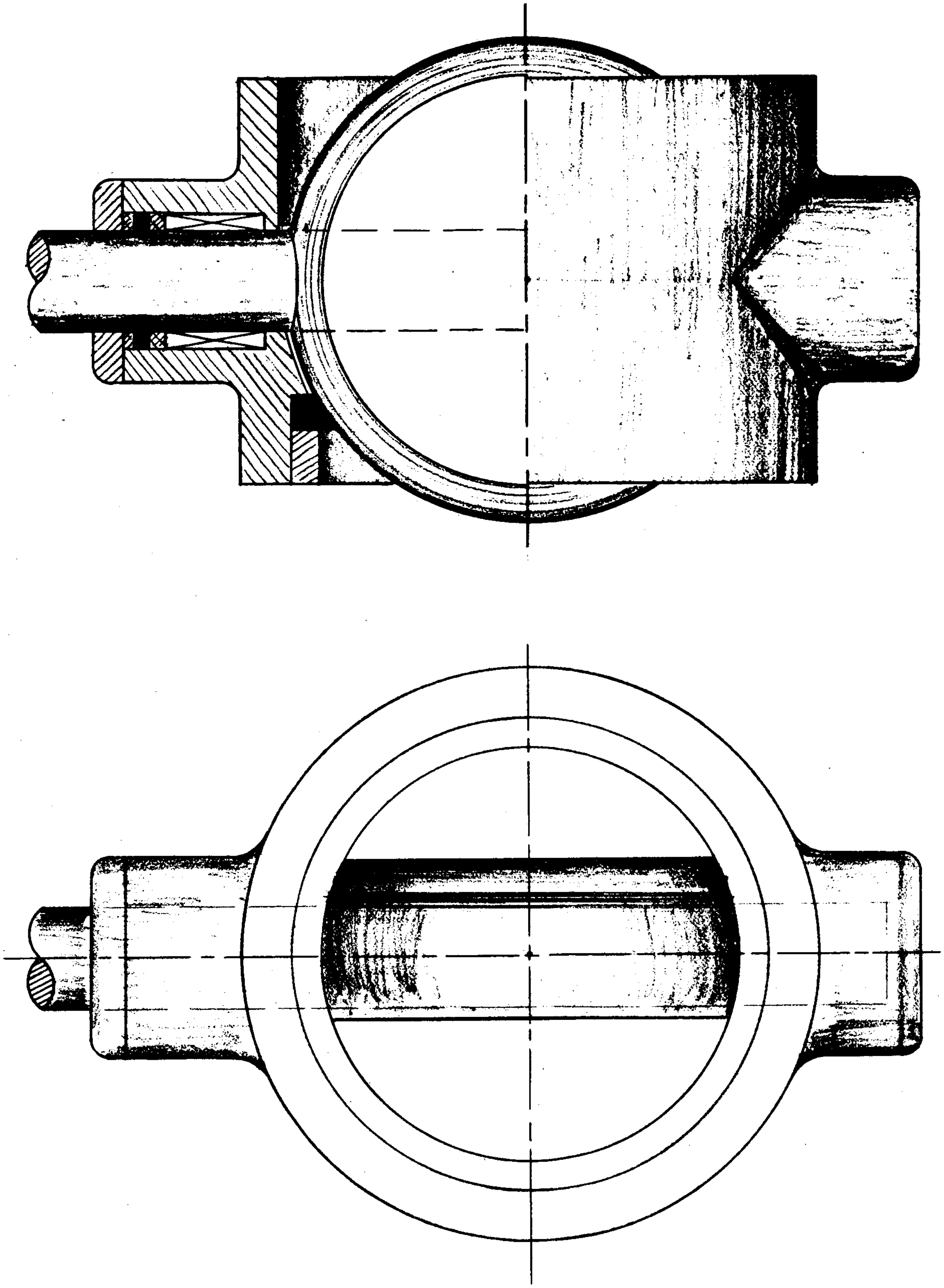
AJ-2 Engine with Venturi Valves

Figure VII-C-12



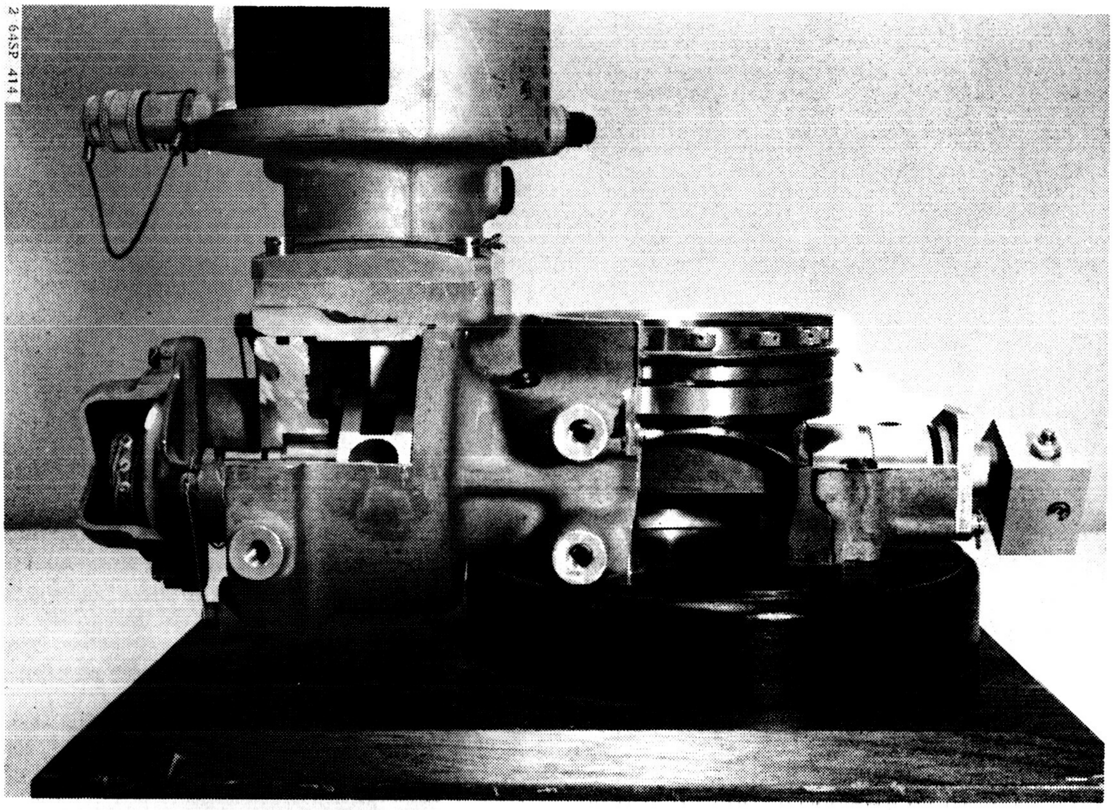
Rotary Sleeve Valve

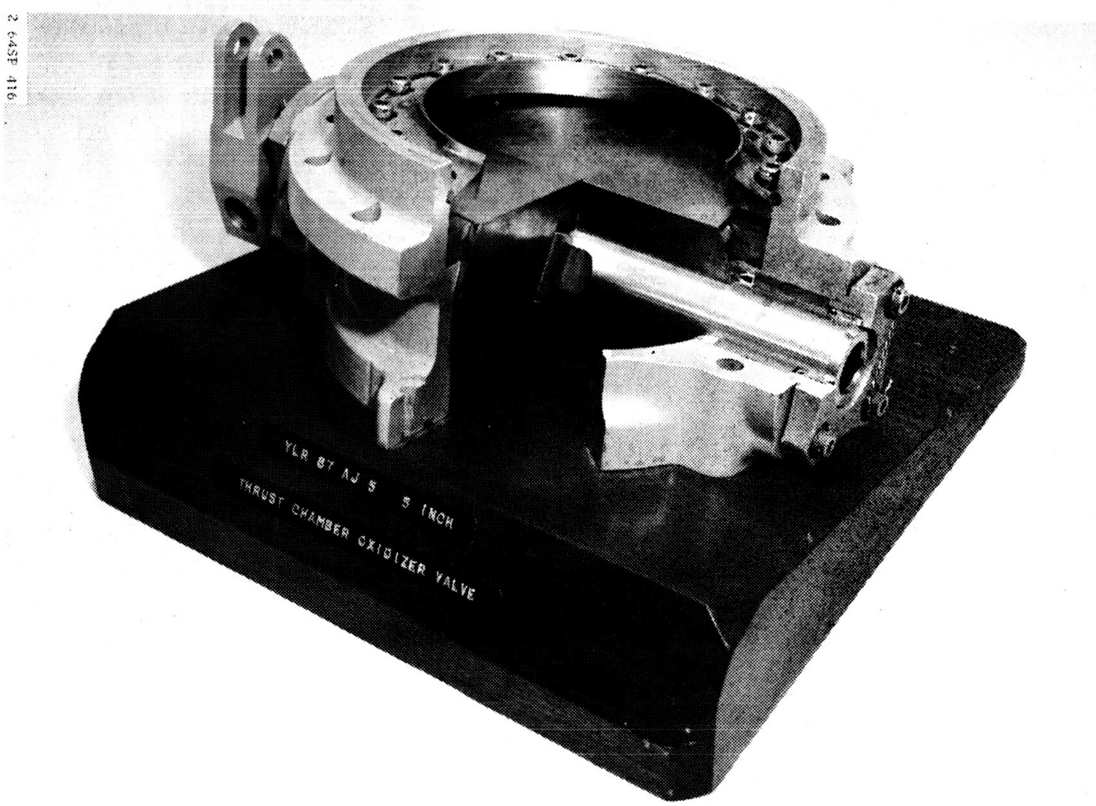
Figure VII-C-13



Butterfly Valve

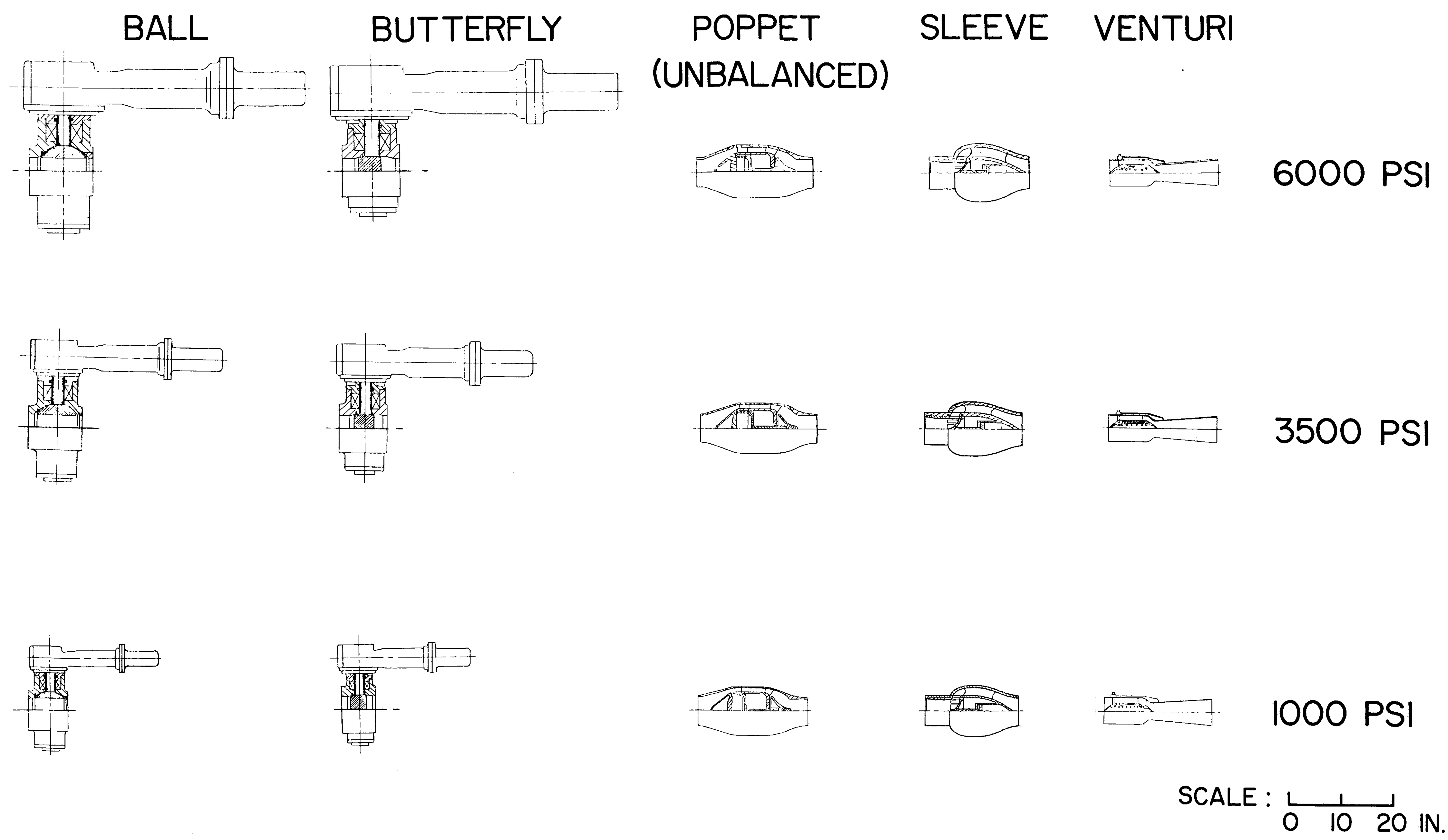
Figure VII-C-14





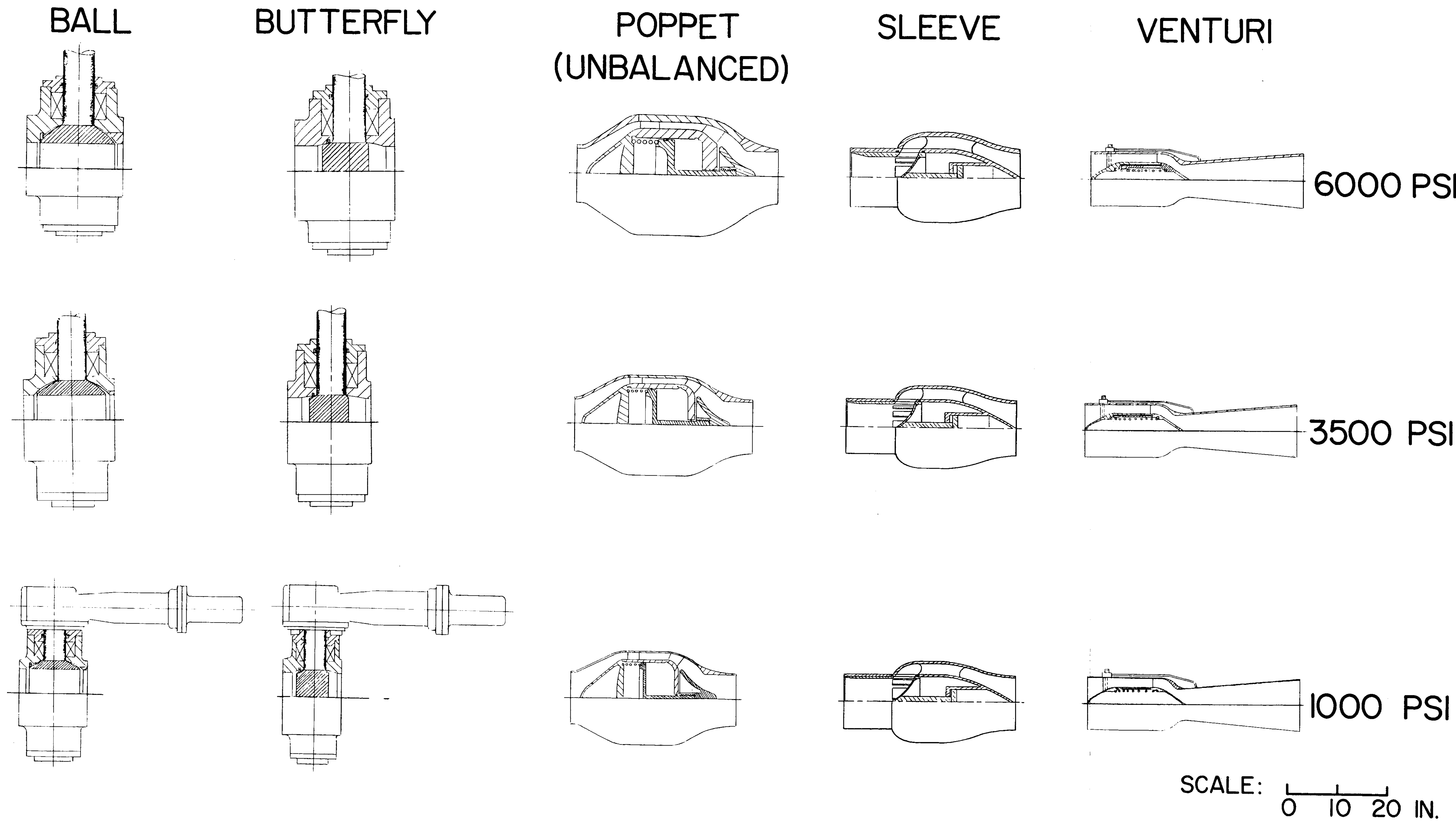
Titan Butterfly Valves

Figure VII-C-15



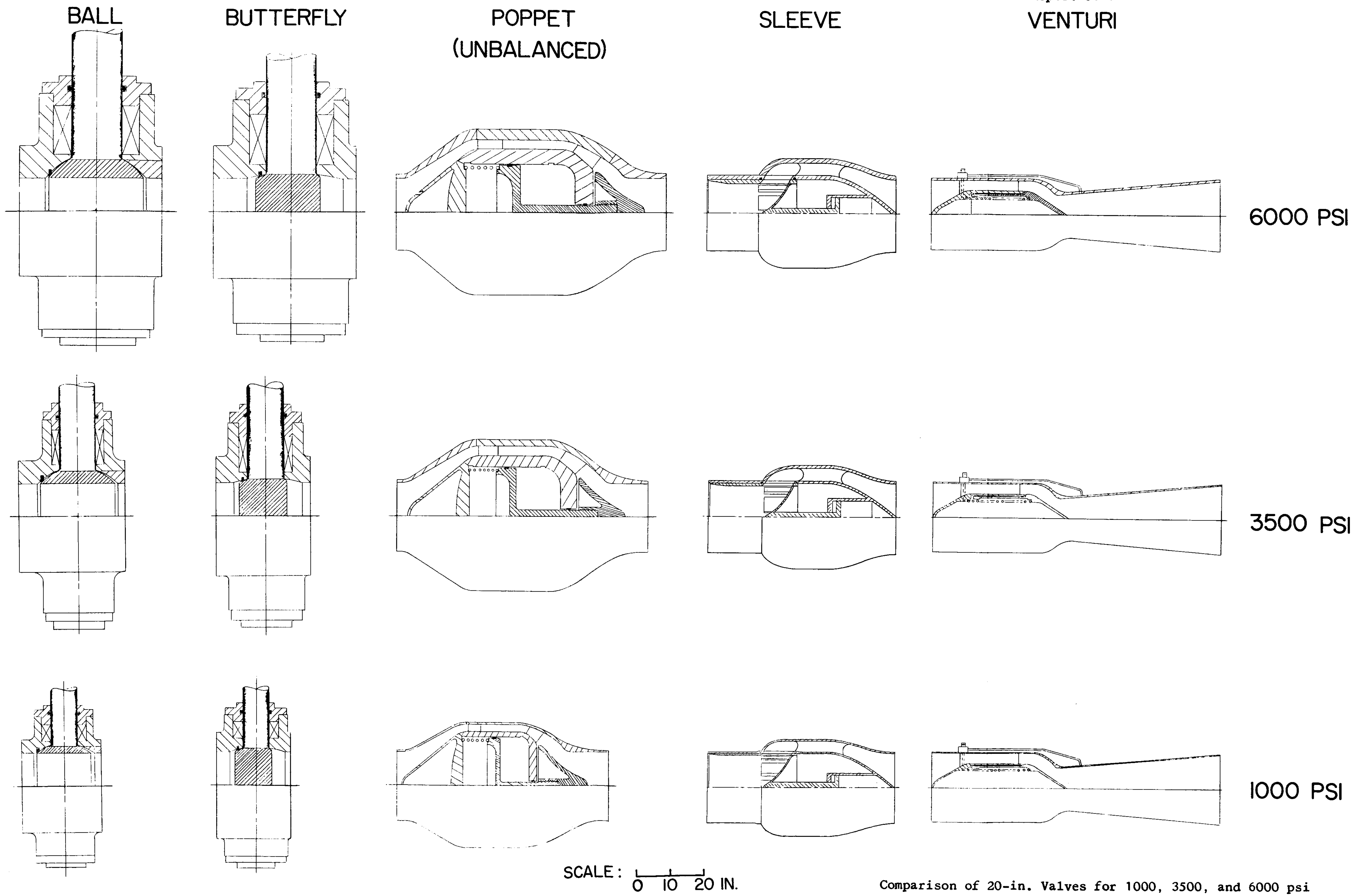
Comparison of 5-in. Valves for 1000, 3500, and 6000 psi

Figure VII-C-16



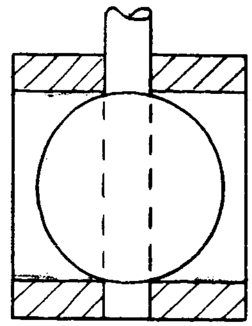
Comparison of 10-in. Valves for 1000, 3500, and 6000 psi

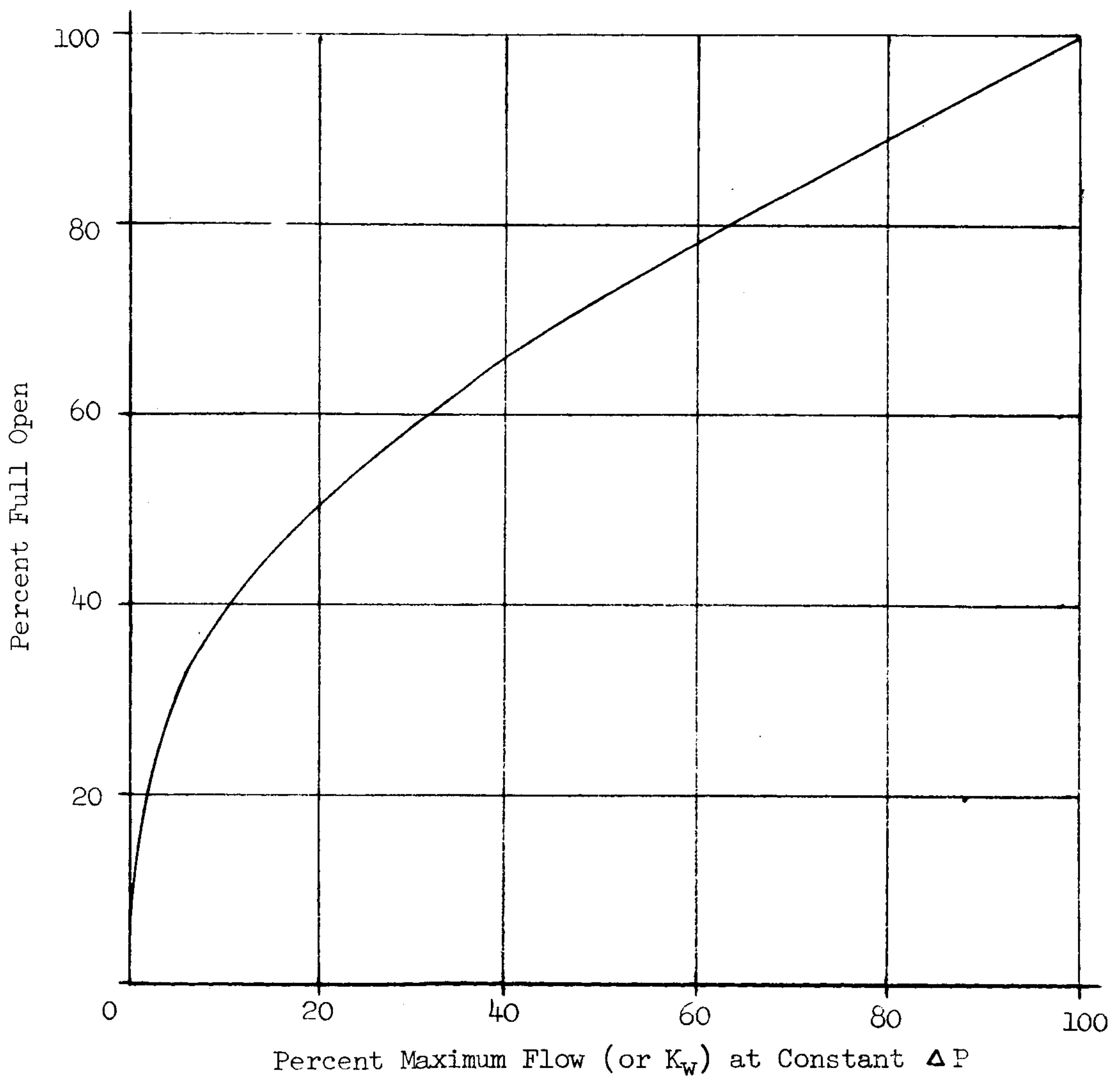
Figure VII-C-17



Comparison of 20-in. Valves for 1000, 3500, and 6000 psi

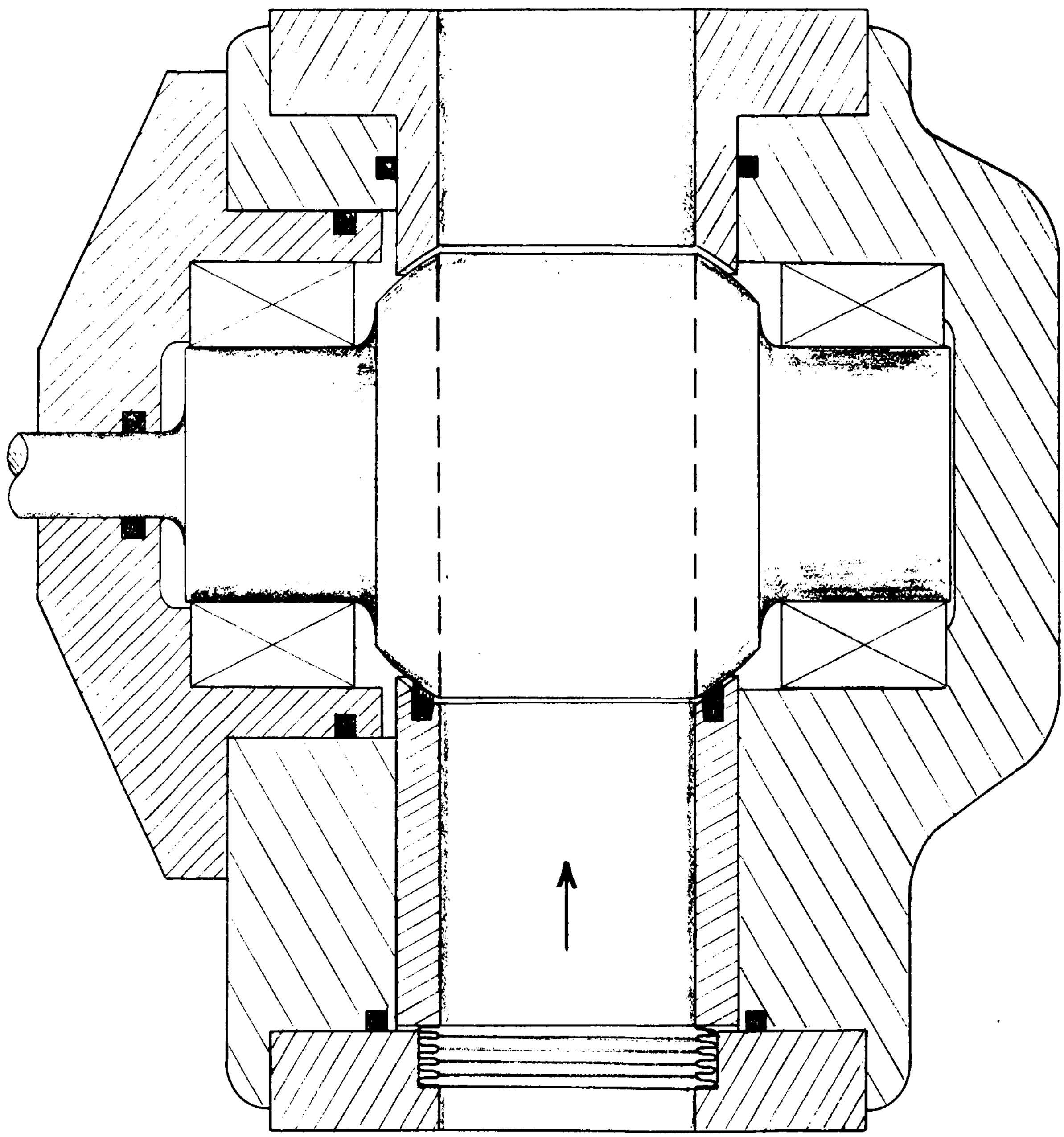
Figure VII-C-18





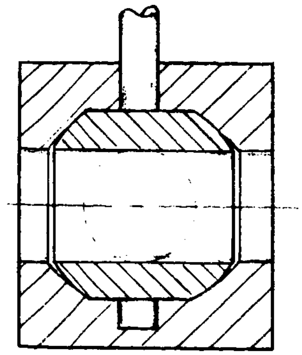
Butterfly Valve Flow Modulating Characteristics

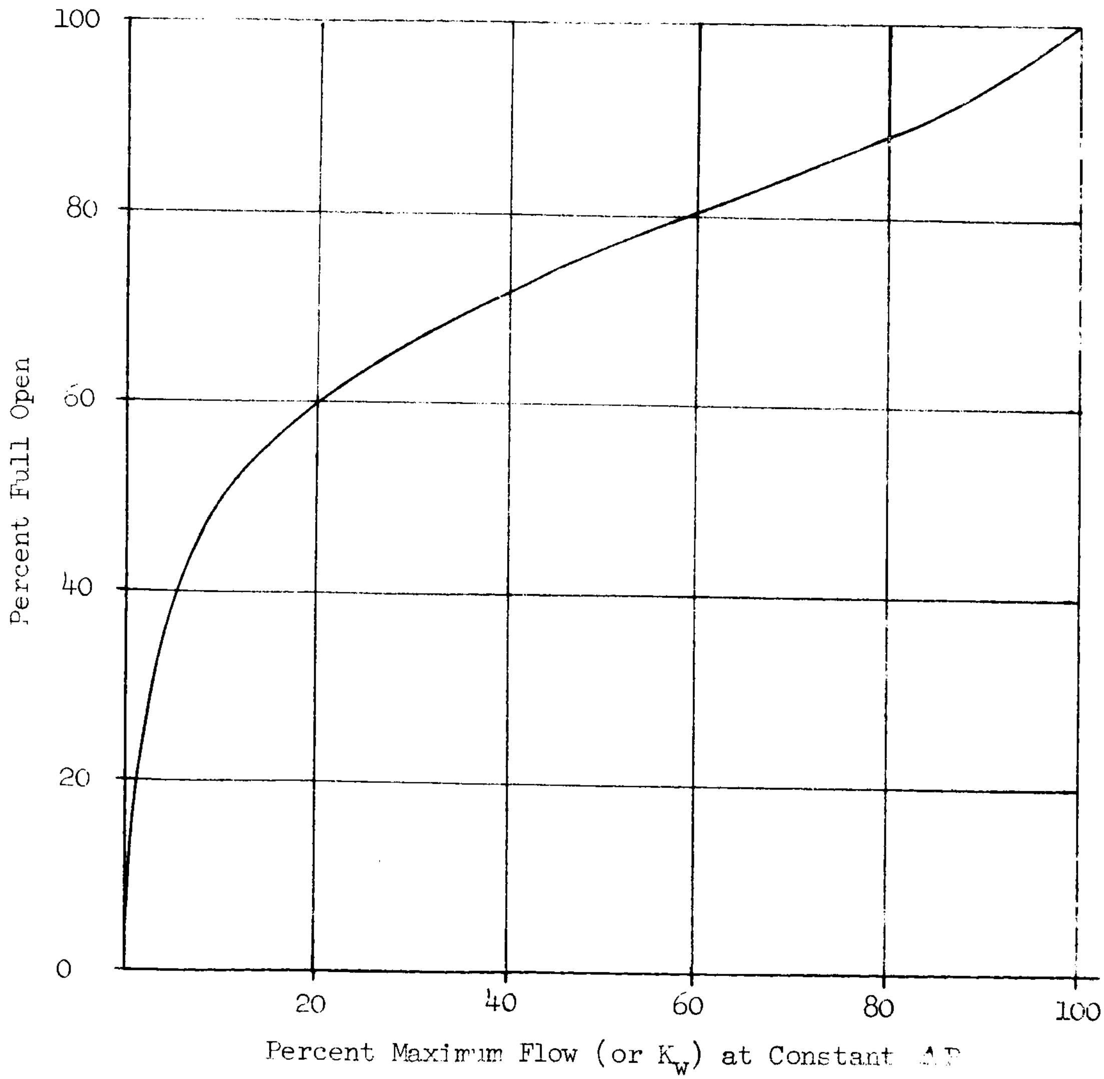
Figure VII-C-19



Ball Valve

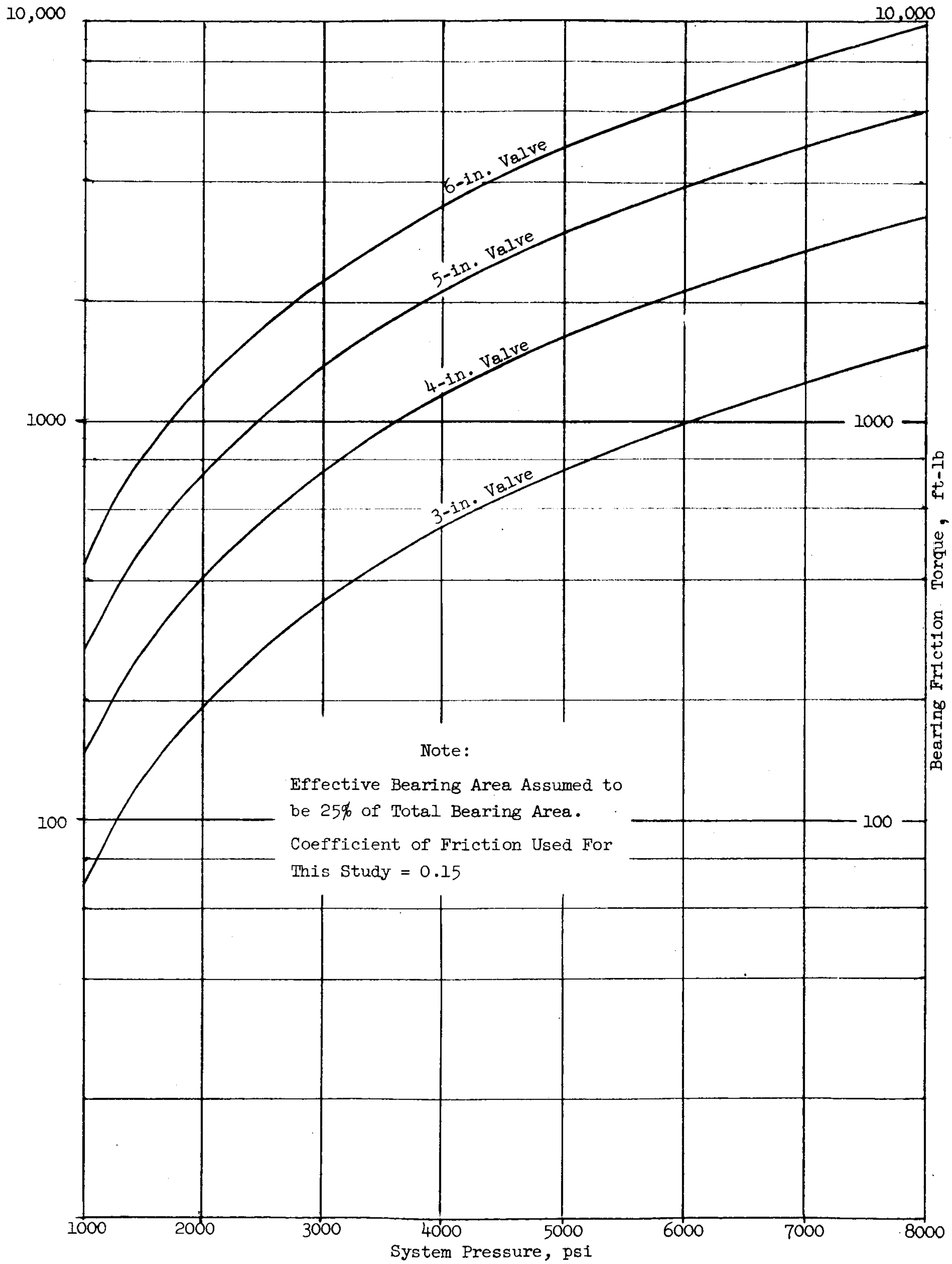
Figure VII-C-20





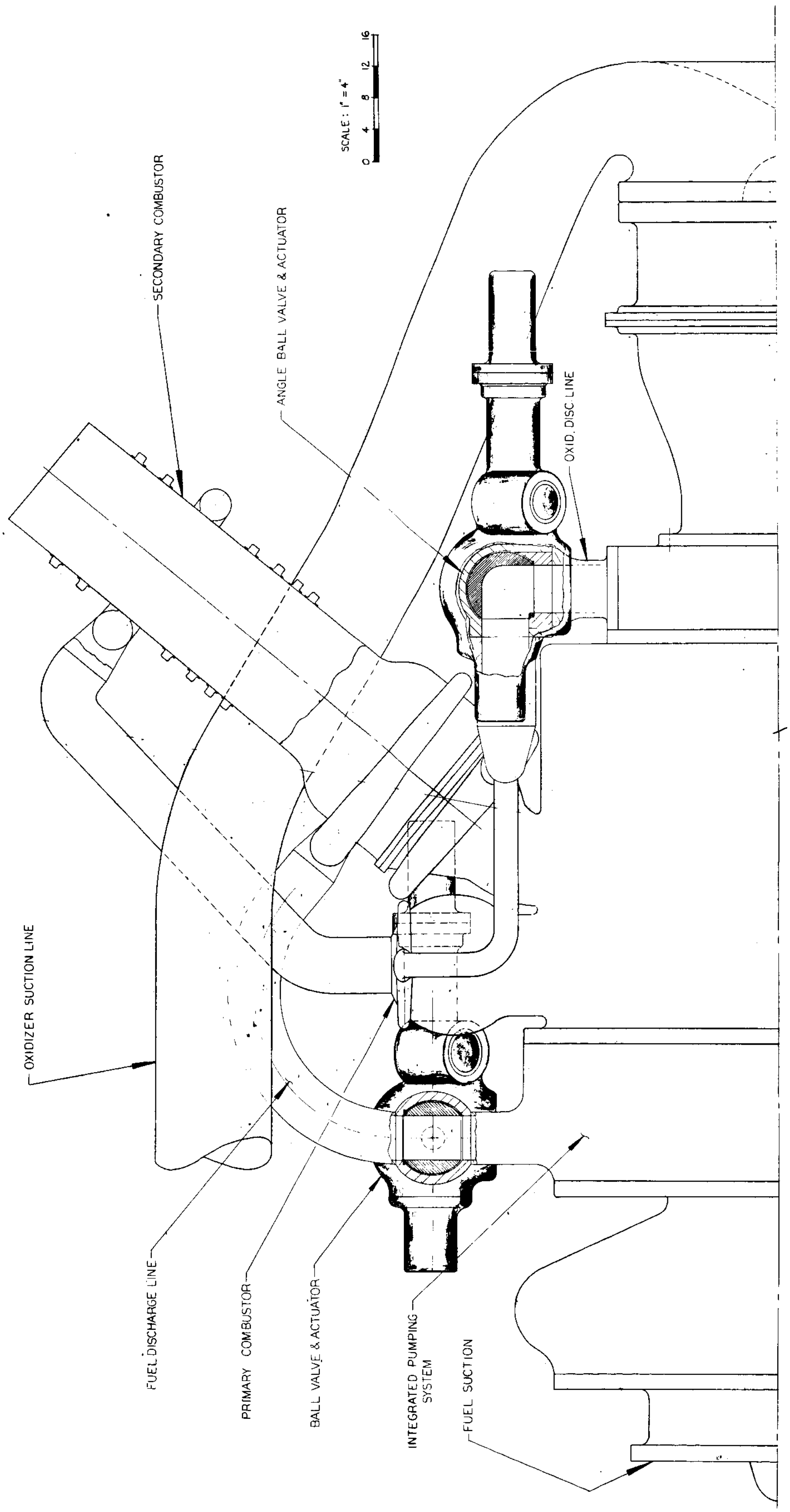
Ball Valve Flow Modulating Characteristics

Figure VII-C-21



Ball Valve Bearing Friction Torque vs Pressure and Size

Figure VII-C-22



AJ-1 Engine with Ball Valves

Figure VII-C-23

VII, Valves (cont.)

D. HIGH-PRESSURE INTEGRAL PUMP DISCHARGE VALVES

1. Ring-Gate Integral Pump Discharge Valves

a. The ring-gate integral pump discharge valve is a pressure-balanced valve because no pressure forces act in the direction of ring motion. The valve consists of a short, large-diameter ring that moves parallel to the pump impeller axis between the impeller discharge and the diffuser section. A principal advantage of this concept is that the flow to multiple discharge lines is controlled simultaneously by a single control element.

b. The total envelope requirement of the integral ring-gate pump discharge valve necessitates a small increase in pump housing diameter. The actuation cylinders may be contained within the ring as is shown in the preliminary design layout (Figure VII-D-1). The compact envelope associated with this type of valve is one of its main features and is effective when space in the pump discharge lines is limited. Stress concentrations that occur at the line-valve connection when a rigid valve is placed in a highly stressed line are not present when this type of valve is used.

c. Seal requirements are the primary limitation to the utilization of this concept. The ring-gate valve requires two large (55 in. for the AJ-1) sliding dynamic seals. Normal clearance for a sliding sleeve of this diameter is a minimum of 0.015 in. Temperature and pressure deflections will require more initial clearance. Extensive disassembly would be necessary to replace the seals (or other valve parts), which limits the choice to seals requiring minimum replacement.

d. The ring-gate pump discharge valve is primarily an on-off type of valve. It can be designed to produce very little effect on the flow stream when fully open. A spring-loaded follower retains the seals and fills the cavity in which the ring-gate seats when closed. The pump impeller-diffuser design must take into

VII, D, High-Pressure Integral Pump Discharge Valves (cont.)

account the radial space required by the valve between these elements. The ring-gate or follower can be designed to incorporate short sections of the diffuser vanes so that no radial discontinuity exists when the valve is open.

e. The placement of the control element in the pump outlet between the impeller and diffuser would have a profound effect on the pump operating characteristics if the valve were used to modulate the high-velocity flow. An alternative would be to employ a vaneless diffuser immediately upstream of the ring-gate valve.

f. One critical problem area associated with this valve concept is the tendency of short cylinders to cock and bind when sliding axially. One solution for this is the use of low-friction ball guides on the ring gate.

2. Multiple-Venturi Integral Pump Discharge Valve

a. This valve concept (Figure VII-D-2) is primarily applicable to engines having multiple discharge lines from the pump or pumps. The positioning of the poppets in the pump discharge ports ahead of the diffuser section allows the use of small valve ports that minimize bore deflection. Since the pump impeller discharge port is the smallest passage area in the system through which all the propellant flows, it is the logical place to locate the poppet seat. The simplicity of the unbalanced poppet and the relative ease with which its opening and closing rates can be controlled make it the logical choice for application to either on-off or modulating pump discharge valve functions. Figure VII-D-3 shows this valve concept applied to the AJ-1 fuel pump.

b. The use of this valve concept to control pump discharge flow results in a slight increase in the size of the pump discharge housing. The diffusers downstream of the poppets require no more space than is needed without the valves; they may be curved if necessary to facilitate line arrangement.

VII, D, High-Pressure Integral Pump Discharge Valves (cont.)

c. The seal requirements for this valve concept are probably the least severe of any of the concepts considered feasible for high-pressure applications. The valve requires only one static seat seal per valve; this seat is much smaller than the main seals that would be required for single line-size valves. The poppet seal diameter of this valve for the AJ-1 engine is only of 3 in. dia, which is within the capability of present-day seals of this type. The poppet to housing clearance does not require a seal in this valve as the small flow through the clearance space can be fed back to the pump suction through the pilot valve circuit.

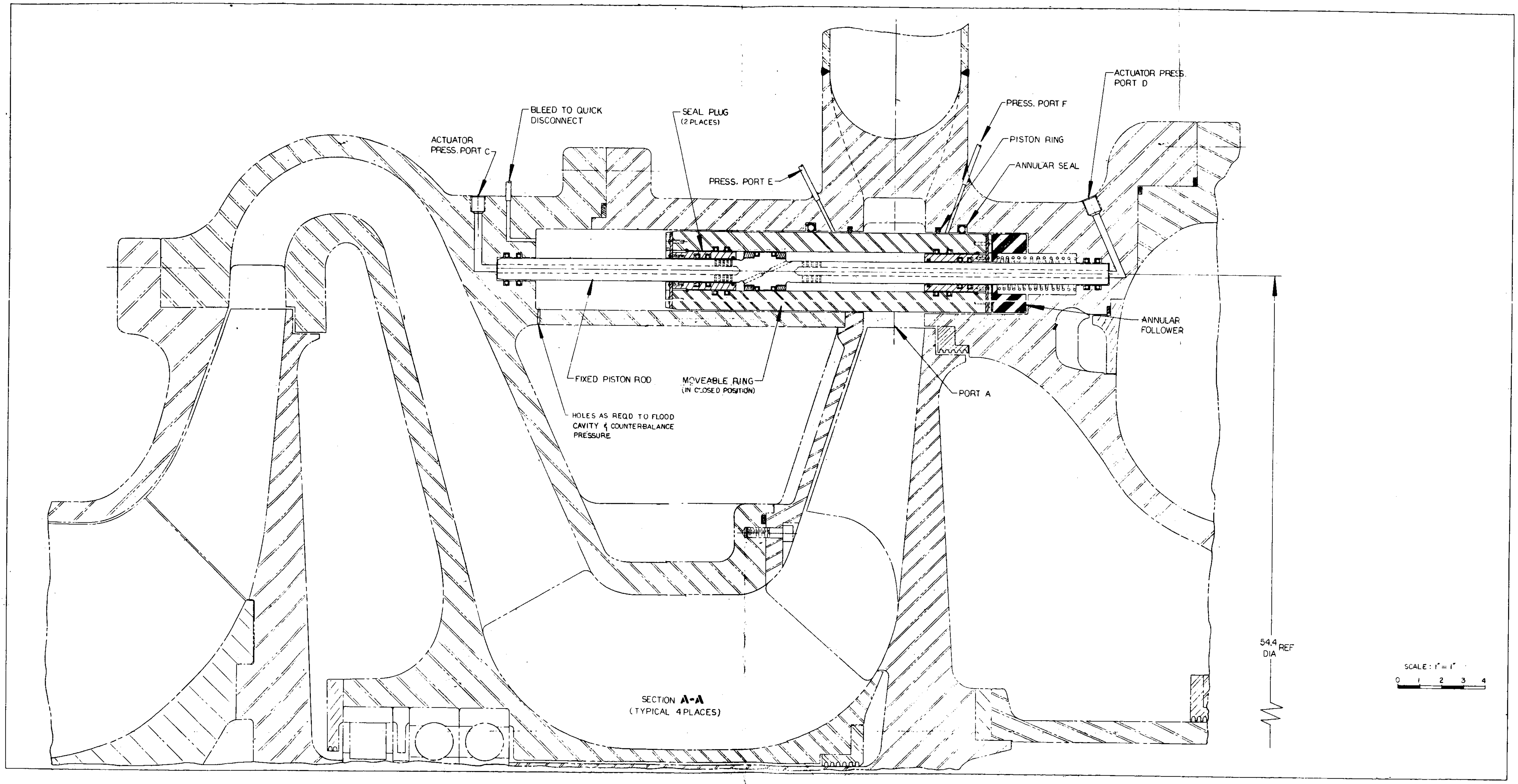
d. By careful streamlining and geometry analysis, the pressure loss from the poppets and their housing can be moderate (Appendix B). The diffuser loss is unaffected by the poppet valves and should not be included in comparing the pressure loss of this valve with other valves.

e. This concept is most applicable when an on-off control is desirable as this greatly simplifies the valve design. As shown in Figure VII-D-2, static system pressure is used to open the valve when pressure behind the poppet is relieved. To give the valve modulating capability, the poppet should be pressure balanced to nullify the effects of fluctuations in pump discharge pressure. An internal piston actuator can be installed within the poppet housing, although a mechanical actuator would be promising for modulating service. The response of this valve would meet the probable requirements of any large engine system. The poppets (the only moving parts) are very small and light compared to other valve types, and considering the pressure forces available, inertial forces will be practically negligible. The response will be primarily dependent on the flow capacity of the pilot valve system. The synchronization of several poppets (eight for the AJ-1) can be controlled to an acceptable degree by utilizing a common pilot valve with equal lengths of line leading to all the valves. Overall engine design considerations could make this approach impractical for specific applications. An alternative is a solenoid pilot valve electrically interconnected for each poppet.

VII, D, High-Pressure Integral Pump Discharge Valves (cont.)

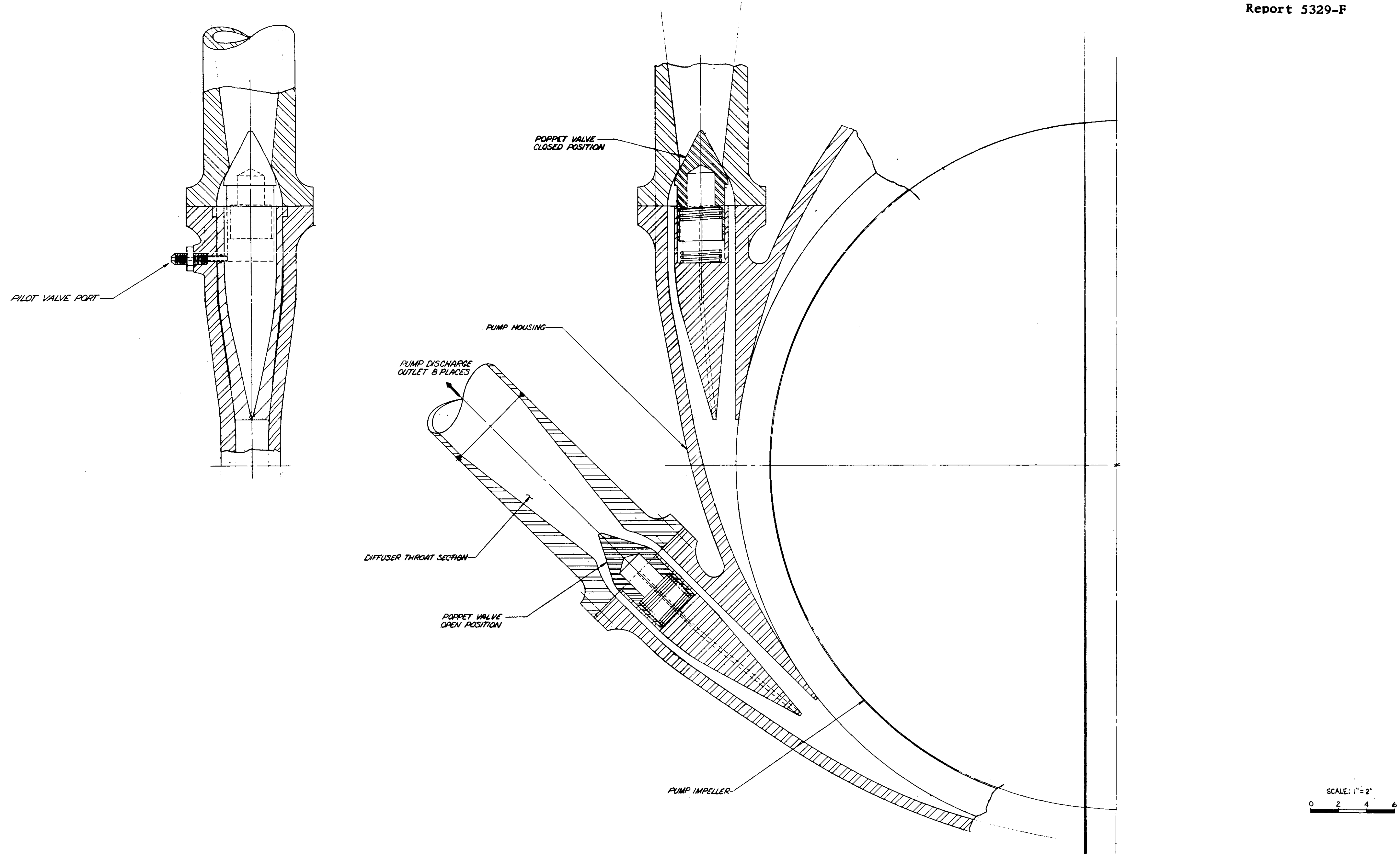
The opening and closing rates of the valves should be very nearly equal since there is no seal friction during actuation.

f. The use of this valve concept is feasible with present seal and design technology for engines similar in size and design (multiple discharge lines) to the AJ-1. However, the effect of this valve arrangement on pump performance has not been analyzed, and development work in this area would be relatively expensive. Obviously, this evaluation has been concerned only with centrifugal pumps. Axial-flow pumps would present different valve design problems. With respect to simplicity, accessibility, and size of functioning parts, this concept is superior to other integral pump discharge valves.



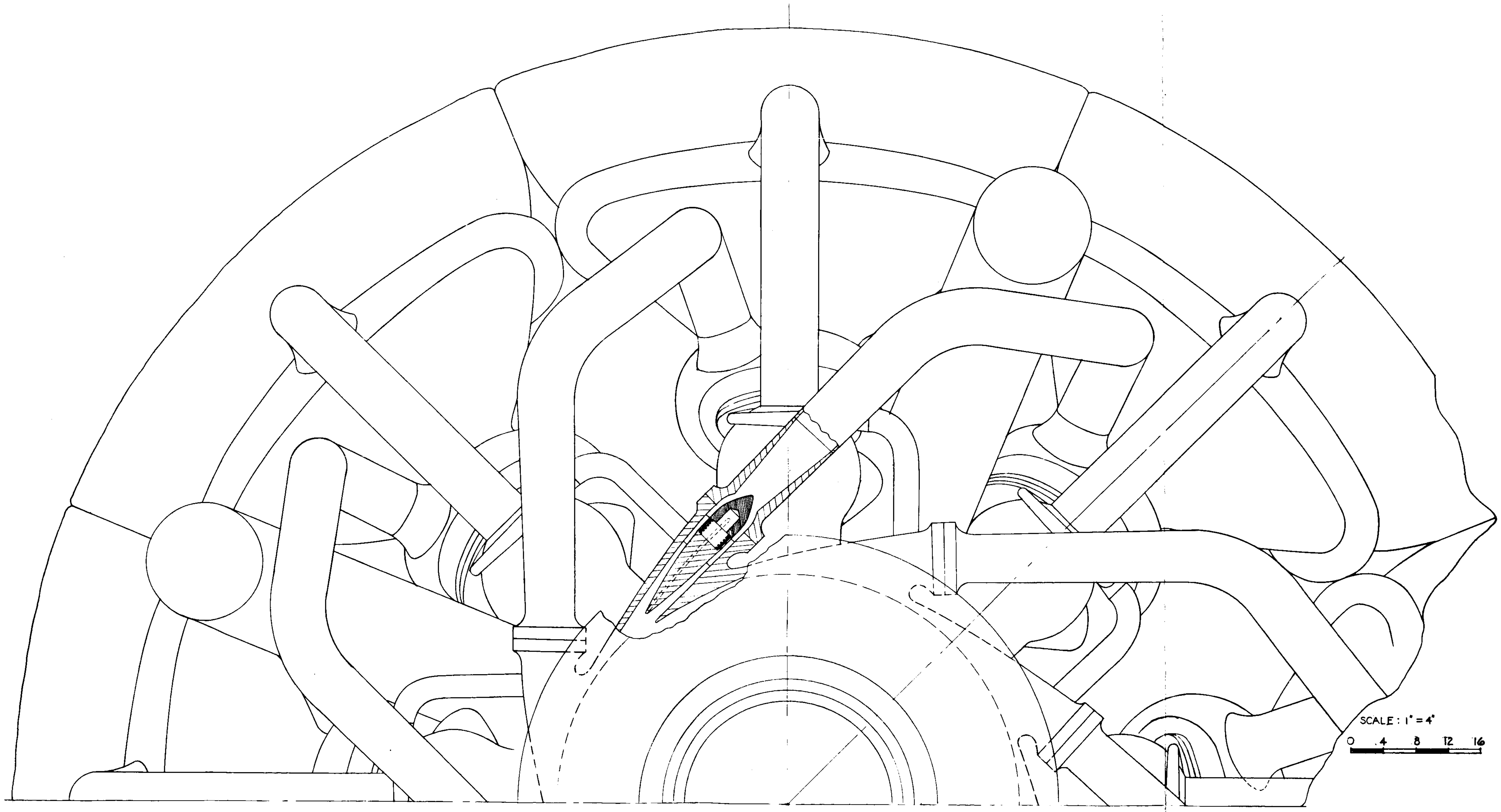
Ring-Gate Integral Pump Discharge Valve

Figure VII-D-1



Multiple Venturi Integral Pump Discharge Valve

Figure VII-D-2



AJ-1 Engine Fuel Pump with Integral Multiple Venturi Valves

Figure VII-D-3

VII, Valves (cont.)

E. LOW-PRESSURE SUCTION VALVES

In selecting a valve concept for a particular pump suction line application, the pressure loss of the valve is important because of the direct effect on the tank pressurization required to supply the necessary pump net-positive suction head. Since the tank wall thickness and weight are directly proportional to the pressurization, high-pressure losses in the suction line can cause significant degradation of the vehicle mass-fraction. Another consideration of primary importance is the valve length. If it must be mounted between the tank bottom and the pump suction port, the length of the valve adds directly to the overall vehicle length. The advantage obtained by mounting the valve within the propellant tank must be weighed against the loss in accessibility. The placement of the valve within the tank has the additional benefit of providing a more stable temperature environment, which is of particular significance for cryogenic propellants. A third area of primary importance is sensitivity to vibration. The large size of the moving element in valves of this size (36 in. dia for the AJ-1) results in high-inertia parts. These parts have to be moved and maintained in position by an actuator operated by an auxiliary energy source, since pump pressure is not available prior to opening of the suction valve.

1. Ring-Gate Suction Valve

a. The ring-gate pump suction valve is well suited for installation within the propellant tank because the valve discharge port can be located on the tank bottom (the low point). The relative sensitivity to temperature differences would be slight since the entire valve and actuator mechanism would be submerged in propellant. Figure VII-E-1 shows such an installation.

b. The envelope dimensions of the internal ring-gate suction valve are not as compact as with most other types of valves, but the entire valve

VII, E, Low-Pressure Suction Valves (cont.)

is within the tank and does not require any space between the tank floor and the pump suction port. The valve mechanism within the tank displaces very little propellant.

c. The seal requirement of the ring-gate (short-sleeve) suction valve will be the primary limitation to the usefulness of this concept. The valve requires two large (approximately 36 in. dia for the AJ-1) dynamic sliding seals. The problem is not as severe as for high-pressure valves where extrusion through a clearance gap is a primary concern, but tolerance of deflection for maintaining a leak-tight seal over such a long circumference is difficult to achieve.

d. With proper design in critical areas such as (1) entrance to the port and (2) shaping of the flow diverter, the pressure loss can be kept to a very low value (essentially the entrance loss to the suction line, Section VI of Appendix B) comparable to the pressure loss of a ball valve.

e. In the event that flow modulation is desired in the pump suction side of the system, the ring-gate valve can readily provide this capability by combining a servo control system with the hydraulic actuator. An electro-mechanical actuator system should also be given consideration in view of the lack of pump discharge pressure prior to the opening of the suction valve.

2. <u>Unbalanced Poppet-Type Suction Valve</u>

a. The desirability of the unbalanced poppet-type valve (Figure VII-E-2) is primarily due to its feature of a single static seal for main poppet sealing.

b. This valve can be designed for mounting completely within the tank (with the poppet stem directed toward the opposite end of the tank) or it can be designed with the poppet shaft located in the center of the pump suction port.

VII, E, Low-Pressure Suction Valves (cont.)

The latter configuration requires a distance of about two-thirds of the port diameter in axial space between the tank bottom and the end of the pump impeller shaft (Figure VII-E-2).

c. Large actuator forces are required for this valve concept. For example, a valve with a seal diameter of 36 in. (AJ-1) and a propellant static pressure of 35 psi requires an actuator force of 34,000 lb to overcome the pressure force. The main poppet seal can be made self-centering, which relieves the guide mechanism of stringent rigidity and tolerance requirements.

d. The valve pressure loss can be kept to a minimum by observing the same considerations mentioned in regard to the ring-gate suction valve. Minimum pressure loss is determined by the practical limitation on the valve stroke. A stroke of about one-third of the line diameter results in a pressure loss approximately equal to 0.5 psi (Section VI of Appendix B).

e. The flow area can be modulated by providing actuator-positioning capability. The shape of the flow deflector can be varied to produce any desired characteristic curve, with the corresponding increase in stroke required to produce the area of maximum flow.

3. Ball- and Butterfly-Type Suction Valves

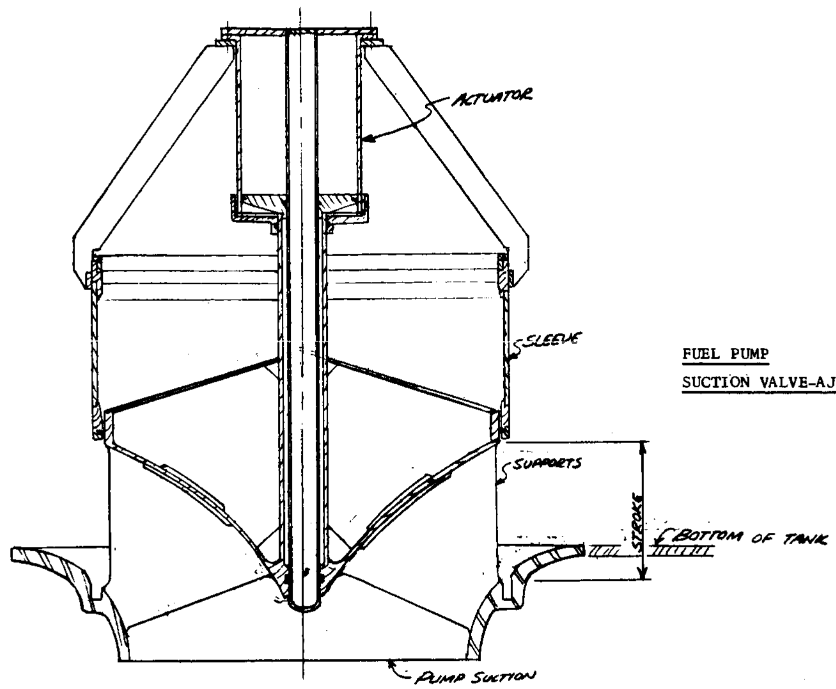
a, b. Ball and butterfly valves are attractive for pump suction applications because of pressure loss considerations.* The nonaxisymmetric pressure loading on the valve housing requires relatively large, heavy housings in order to maintain seal geometry. The large, nonsymmetrical moving elements are also conducive to seal problems unless the parts are made relatively heavy in comparison with other types of valves. In considering the ball valve for a single suction line between the tank and the pump suction port, the valve cannot be placed within the tank because the valve inlet would be above the tank low point. This necessarily requires a sizable mounting space that would add to the overall length of the vehicle.

c. Seal requirements of both ball and butterfly suction valves have been satisfactorily met for existing suction valve applications.

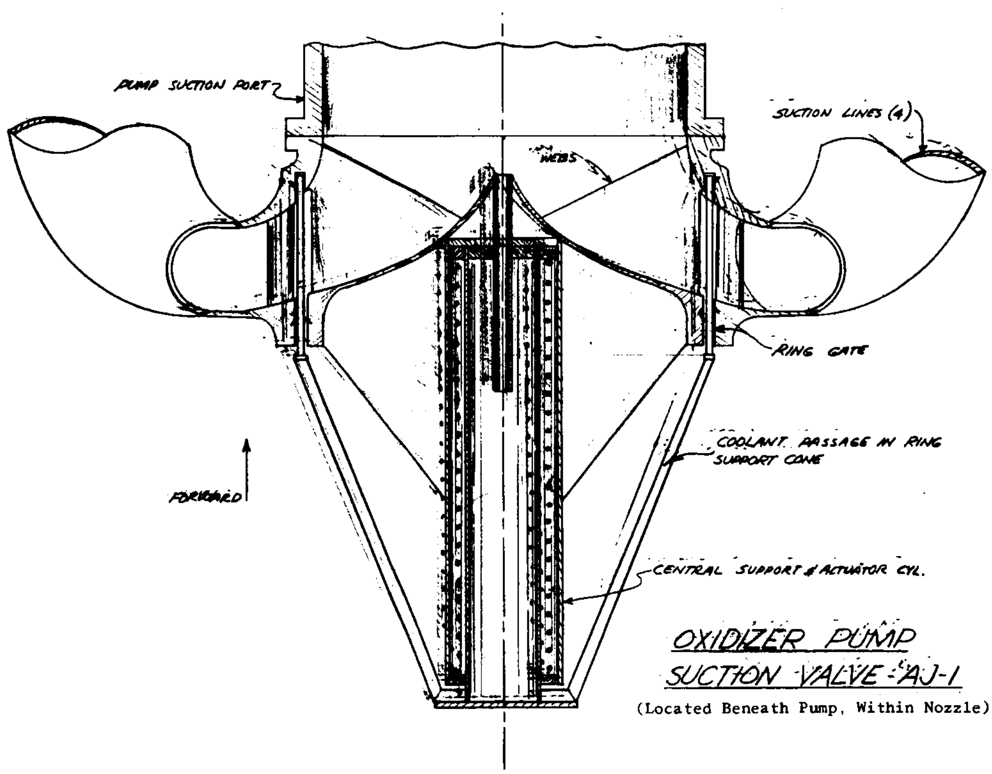
d. The pressure loss for ball valves is almost equivalent to an equal length of straight line. Butterfly valve pressure drops are appreciably higher (Table VII-C-1) and are a function of the design pressure.

e. These valve types require external actuators to supply an amount of torque corresponding to the circumferential seal length and pressure (Appendix F and Figure VII-C-22). In addition, the butterfly valve experiences high torque because of dynamic flow forces on the blade near the fully open position.

*Ball valves are currently in use in pump suction lines on the Saturn vehicles; butterfly valves are being used as the prevalves in the suction lines on the Titan Missile.



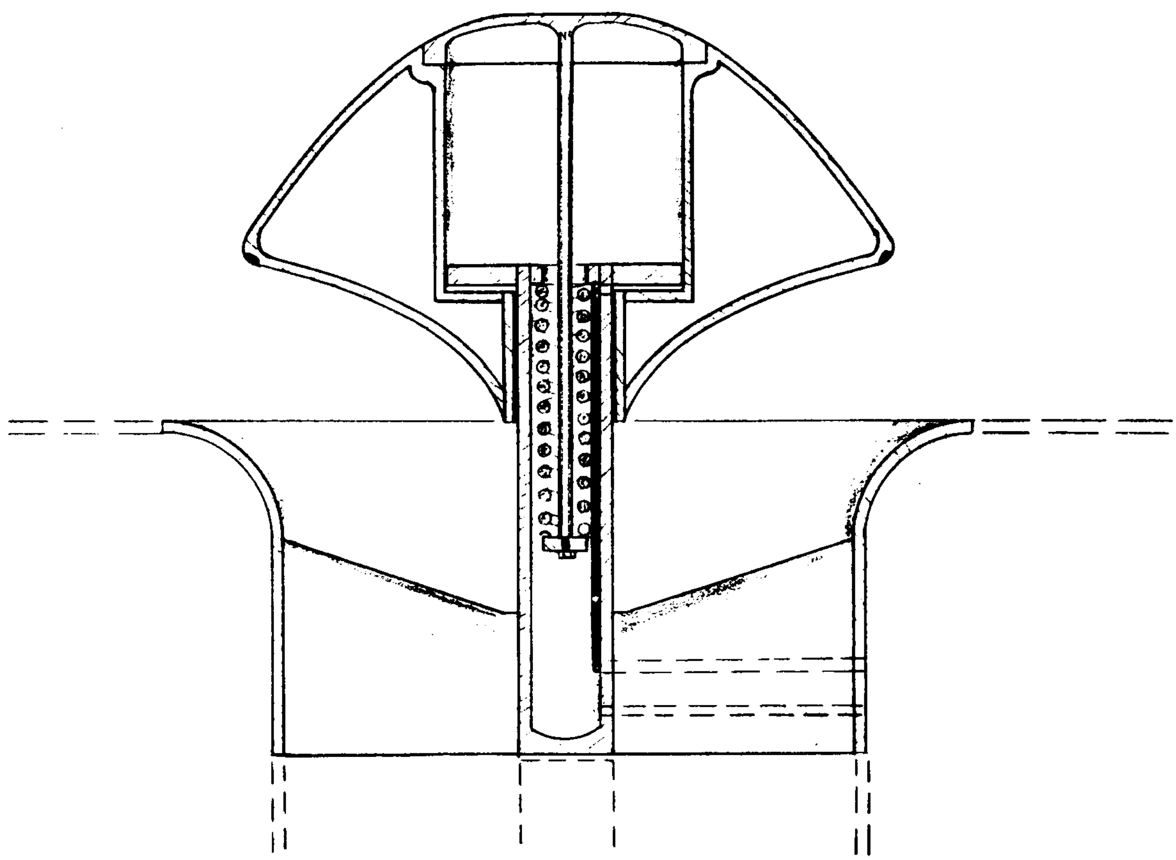
FUEL PUMP
SUCTION VALVE-AJ-1



OXIDIZER PUMP
SUCTION VALVE-AJ-1
(Located Beneath Pump, Within Nozzle)

Ring-Gate Pump Suction Valves

Figure VII-E-1



Unbalanced Poppet Suction Valve

Figure VII-E-2

VII, Valves (cont.)

F. VALVE DESIGN PROBLEMS AND REQUIRED TECHNOLOGY

The most critical problem associated with fluid-control valves for liquid-rocket engines is sealing. The extrapolation of present valve-design philosophy to large size, high-pressure application will result in extremely heavy valves with very high actuator-power requirements. The pressure-balanced valve concepts, which are best suited to large, high-pressure applications, present sealing problems that are beyond the present state of the art.

The design of high-pressure flow-control valves for liquid-rocket engines require treatment of four basic areas:

1. Sealing (internal and external)
2. Actuation
3. Resistance or pressure drop
4. Structural integrity.

The latter three aspects of valve design are relatively straight-forward, but are dependent to some degree on the sealing technique employed. Therefore, while sealing is the primary valve-design problem, it cannot be totally divorced from other valve-design considerations and treated independently. For example, sleeve-valve actuator-power requirements will increase with pressure if pressure-loaded seals such as lipseals are used, whereas metal piston rings would provide a friction drag relatively independent of pressure. On the other hand, metal piston rings would dictate a very stiff valve body to maintain close tolerances on roundness, while elastic lipseals can tolerate larger changes in cylinder shape. The very high reaction loads imposed by high-pressure lines or ducts will add to the distortions caused by thermal expansion and the pressure within the valve, to further complicate the sealing problem. If the optimistic assumption is made that bellows joint spring rates for high-pressure lines will increase proportionally to the increase in pressure, the reaction loads imposed on a 6600-psi valve will be three to four times greater than those imposed on existing valves, such as the M-1 sleeve valves.

VII, F, Valve Design Problems and Required Technology (cont.)

The advanced-technology work on static, sliding, and rotating seals, which is presently being conducted by various agencies under NASA and Air Force direction (Appendix A), should yield the required information concerning seal behavior at the sealing surface. The present scope of these programs does not permit answering the basic questions of how to design large valves that will satisfactorily seal high pressures. A program is needed that will apply new seal technology to the valve-design problems and thereby provide the needed information for designing specific large high-pressure valves. Such a study should include sleeve and poppet valves for 6600-psi operating pressure at sizes from 8- to 20-in. nominal diameter. Adequate means should be shown for providing modulating capability over the entire range of valve lift, with emphasis on the seal requirements imposed by such modulation.

VII, Valves (cont.)

G. RECOMMENDED ADVANCED TECHNOLOGY VALVE PROGRAM

The purpose of this recommended program is to apply the results of recent comprehensive seal studies to the basic design requirements of large, high-pressure, rocket engine valves in order to demonstrate the feasibility of those desirable valve concepts whose application has been limited by the lack of adequate seal technology. The information now being compiled under contracts such as those listed in Appendix A should provide sufficient information relative to static seals, sliding seals, rotating seals, and seats and poppets to either (1) permit the design of large (8- to 20-in.), high-pressure (up to 6600 psi) sleeve and poppet valves or (2) provide criteria for establishing limits for the application of these valve types. This study should emphasize means of establishing minimal leakage (zero leakage, or 1×10^{-3} cc/sec, if possible) at high pressure for both cryogenic (to -423°F) valves and hot-gas (to 1500°F) valves.

The actuator should be capable of positioning the poppet or sleeve at any desired percent of stroke while the unit is under pressure. It is recommended that mechanical or electro-mechanical actuation be evaluated as well as fluid-pressure actuation, to demonstrate the desirability of low seal friction and pressure-balanced design.

Particular attention should be directed to the valve structural and actuator requirements dictated by the particular seal concepts as well as any effects the seal may have on pressure-drop performance. Enough experimental data should be obtained and sufficient analysis performed to permit the selection of valve concept, seal type, actuator, and materials for a particular high-pressure valve application without the necessity of a long preliminary seal-development program such as those that preceded the development of high-pressure valves on many present-day rocket engines.

VII, G, Recommended Advanced Technology Valve Program (cont.)

It is recommended that the final experiments in this program include at least some full-scale testing to fully evaluate the combined effects of size, pressure, clearance, materials, and tolerances on leakage and actuator requirements. It is visualized that a single valve body could be designed to accommodate interchangeable seals, seats, sleeves, poppets, and actuators, thereby providing a consistent point of reference while minimizing expenditure for test hardware.

VIII. LINES, DUCTS AND FLEXIBLE JOINTS

A. FLUID-TRANSFER CONSIDERATIONS

The design requirements for fluid-transfer lines in rocket-engine systems is dictated by engine-systems constraints, such as fluid physical properties, fluid pressure, allowable pressure drop, end reactions, end relative movement, and line envelope. Some of the line-system interfaces and interactions may be varied to achieve optimum design; however, other system parameters, such as fluid properties and pressure, are specified for a particular engine system.

The working fluids of large high-pressure liquid rockets that normally require fluid-transfer lines fall into three general categories: (1) low-pressure propellant in suction or feed lines, (2) high-pressure propellant downstream of the pump, and (3) high-pressure hot-gas products of combustion downstream of the gas generator(s) or primary combustor(s). While the primary emphasis during this program has been on those components affected by high pump discharge pressures, the relatively low-pressure (30 to 650 psi) suction lines have also been studied. This was done to evaluate the relative severity of the demands of high flow-rate alone, as compared with those of high flow-rate combined with high pressure. Table VIII-A-1 summarizes the key functional parameters for the various lines analyzed.

The large single-pump engine configurations utilizing multiple combustion chambers require relatively small suction lines (not over 19-in. dia) to fit the lines between the combustion chambers. Figure V-B-1 illustrates this concept for engines AJ-1 and AJ-2. For an engine configuration of this type, suction-line diameters for the 24 x 10^6-lb-thrust vehicle will probably be comparable to the 17- to 19-in. lines of the present Saturn vehicles. For the same fluid velocity as the AJ-1 (24.4 ft/sec), a single suction line for the AJ-6 would have a diameter of 27-in.

The most significant suction-line design requirement of the large-thrust booster is the increased pressure requirement. The vehicles evaluated under this

VIII, A, Fluid-Transfer Considerations (cont.)

program would require oxidizer suction lines up to 136 ft high. This static head of LO_2 under 1-g loading is equivalent to 66.5 psi at the pump inlet, 665 psi at burnout with an acceleration of 10 g; and 399 psi if the acceleration is only 6 g. By comparison, present Saturn vehicles require proof pressures for suction-line components in the vicinity of only 200 psi.

The high-pressure propellant lines present more severe problems than the suction lines because the length-to-diameter ratio is usually much lower (3:1 to 20:1 compared with up to 130:1 for suction lines), and the choice of line geometry is limited by the location of the primary engine components and/or the necessity to include a valve in the line.

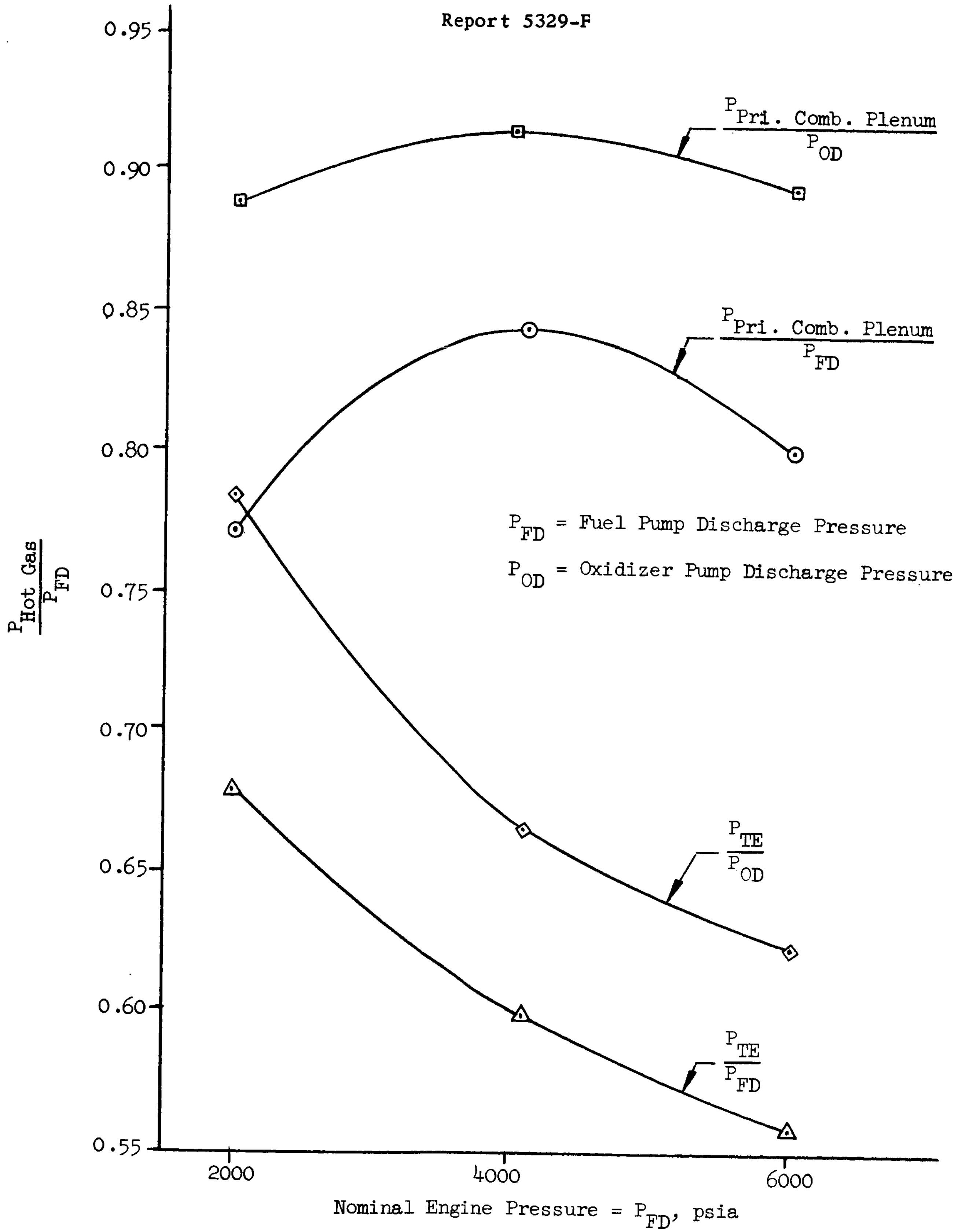
The design of hot-gas ducts presents severe line-design problems for these engines. While the pressures involved are somewhat lower than the high-pressure propellant, ranging from 55 to 90% of nominal pump discharge pressure, P_D, (Figure VIII-A-1), the high temperature (700 to 1900°R) reduces the allowable stress levels of the line materials. In addition, the dimensional changes resulting from a temperature rise of 1400°F are proportionately greater than those of a cryogenic line with temperature changes of 500°F.

The following section discusses the various considerations associated with the design of ducting systems for high-pressure engines, the problems involved, and the available or potential approaches to solve these problems.

TABLE VIII-A-1

SUMMARY OF FLOW PARAMETERS FOR REPRESENTATIVE LINES

Engine	Line Description	Line Fluid	Pressure One-G, psi	Pressure 10-G, psi	Operating Temp, °F	ID in.	$\dot{w}$, lb/sec	Total ΔP One-G, psi	Total ΔP 10-G, psi	Figure No. VIII-B
AJ-1	Fuel line to primary combustor	Liquid H_2	4100	4900	-210	6.5	278	22.4	29.5	1
	Fuel line to secondary combustor	Liquid H_2	4550	5350	-423	5.0	278	21.9	20.8	2
	Fuel line to secondary combustor with valve	Liquid H_2	4550	5350	-423	5.0	278	20.0	18.6	3
	Oxidizer line to primary combustor	Liquid O_2	4100	4900	-293	4.0	251	15.2	23.2	4
	Oxidizer suction line (4/Eng)	Liquid O_2	53		-293	17.8	3300	-56.0	-609.0	5
	TVC line	Comb. Prod.	3500		1440	7.0	628			
AJ-2	Fuel line to primary combustor	Liquid H_2	4100		-403	8.0	740	22.0	23.4	6
	Fuel line to secondary combustor	Liquid H_2	4700		-413	8.0	740	17.4	16.0	7
	Oxidizer line to secondary comb.	Liquid O_2	4100		-290	8.0	3825	38.4	53.5	8
	Fuel line to primary combustor	Liquid H_2	4100		-403	8.0	740	14.7	14.7	9
	Oxidizer suction line (6/Eng)	Liquid O_2			-293	18.8	9000	-14.0	-614.0	10
AJ-3	Fuel line to primary combustor	RP-1	3900		100	1.5	47	22.4	29.2	11
	Fuel discharge line	RP-1	4000		80	4.0	640.5	41.5	34.6	12
AJ-5	Fuel pump discharge line to combustion chamber manifold	LH_2	4075		-410	6.75	742	48.0		13
	Fuel pump discharge to inj	LH_2	4075		-410	7.5	1446	298.0		14
	Oxidizer discharge line	LO_2	4075		-290	17.0	13,127	45.0		15
	Turbine hot gas line									
	Single line	Comb. Prod.	540		690	17.9	512	10.5		16
	Double line	Comb. Prod.	540		690	12.7	512	12.3		17
AJ-6	Fuel line to primary combustor	LH_2	3960		-210	10.0	1119	28.8		18
	Oxidizer line to primary combustor	LO_2	4000		-290	4.75	1024	50.0		19
AJ-8	O_2-rich hot gas line to injector	Comb. Prod.	3000		1062	12.8	1567	20.0		20 and 21
	H_2-rich hot gas line to injector	Comb. Prod.	3000		1059	12.3	380	20.0		22



Ratio of Hot Gas Pressure to Nominal Engine Pressure

Figure VIII-A-1

VIII, Lines, Ducts, and Flexible Joints (cont.)

B. DESIGN OF LINES

As discussed in the previous paragraphs, the line design is dependent on the fluid, the environment, and the line geometry. This portion of the discussion will serve to explain how descriptive parameters such as diameter, wall thickness, material properties, and line configuration may be varied to meet the fluid and environmental requirements. The manner in which sample lines were analyzed to establish design criteria is shown, along with the actual results.

1. Fluid-Flow Line Considerations

The only fluid-flow consideration in lines that has an appreciable effect on engine-system performance is the pressure drop of the fluid as it passes through the line. The steady-state engine-system analysis indicates that the sensitivity of engine performance to this pressure drop is a function of the engine cycle, nominal engine-system pressure, and the location of the line in the engine system. For example, the line providing oxidizer to the secondary injector plenum of a high-pressure staged-combustion cycle engine, such as AJ-1, is allowed a pressure drop of several hundred psi and may use an orifice to obtain the decrease.* The low-pressure version of the AJ-1 engine, the AJ-2000, requires a pressure drop of 268 psi as compared with the 1152 psi of the AJ-1 baseline engine that operates at approximately twice the system pressure. A gas-generator cycle engine, such as AJ-5, has no line where such a high pressure drop is required.

Twenty-one different lines (see Section VI), each selected because it represented a particular problem, were analyzed to determine pressure drop as a function of line diameter. The results are shown in Figures VIII-B-1 through -22 and the selected diameters and pressure drops are tabulated in Table VIII-A-1.

*The pressure drop in this instance is required so that the oxidizer may enter the secondary combustor injector at a pressure comparable to that of the fuel-rich hot gas which has lost pressure through the primary combustor and turbine before reaching the secondary combustor.

VIII, B, Design of Lines (cont.)

Entrance and exit losses were not considered for any of the lines. These losses were considered to be part of the pressure drop associated with the major component (pump, plenum, etc.) to which the lines is mated.

The method used to calculate pressure loss due to flow in a line may be found to vary somewhat, but the simple approach used in this analysis is essentially that presented by the Crane Company* using Darcy's equation, which follows:

$$\Delta P = f \frac{L}{D} \frac{V^2}{2g}$$

where: ΔP = Pressure drop, psi

f = Friction factor, a function of Re, $\frac{\epsilon}{D}$

$$R_e = \frac{VD\rho}{\mu}$$

where:

R_e = Reynolds No. (dimensionless)

V = Velocity, ft/sec

D = Inside diameter of line, in.

ρ = Fluid density, lb/ft^3

μ = Viscosity, lb/ft-sec

ϵ = Roughness factor (inside line wall), in.

g = Gravitational constant, 32.17 ft/sec^2

Because of the relatively small loss coefficients $(F, \frac{L}{D})$ encountered in the engine systems under consideration, the compressible fluids were treated as incompressible for velocities less than Mach 0.2.** Hence, the above expression was used for gas as well as liquid pressure-drop analysis.

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*Technical Paper 410, Flow of Fluids Through Valves, Fittings and Pipe, Crane Company, Chicago, 1952.

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VIII, B, Design of Lines (cont.)

Bends and elbows in the line are translated into equivalent lengths of straight line. In practice, these equivalent lengths vary with diameter, particularly in the case of short-radius bends, the radii of which approach one-half the line diameter. An interesting example of this is shown in the pressure-drop curve for the oxidizer line to the primary combustor of the AJ-1 engine (Figure VIII-B-4) or the fuel-pump discharge line to the injector of the AJ-5 engine (Figure VIII-B-14). In some cases, the added length of line necessary to achieve gentle bends results in negligible improvement in pressure drop. An example of this may be seen in the two alternative designs for the AJ-8 oxidizer-rich hot-gas duct (Figures VIII-B-20 and 21). For a 20-psi pressure drop, the change from two 90° ten-inch-radius elbows to a single 180° thirty-one-inch-radius bend permits less than 4% reduction in line diameter. The pressure loss versus diameter curves, Figures VIII-B-1 through -21, were calculated by the continuity and Bernoulli equations for uncompressible flow.

$$Q = A_1V_1 = A_2V_2$$

$$\frac{P_1}{\rho_1} + \frac{V_1^2}{2g} + Z_1 = \frac{P_2}{\rho_2} + \frac{V_2^2}{2g} + Z_2 + H_{losses}$$

For $V_2 = V_1$

$\rho_1 = \rho_2$

$$P_1 - P_2 = \left[(Z_2 - Z_1) + H_{losses}\right]\rho$$

where:

1 = Inlet

2 = Exit

$P_1 - P_2$ = Pressure differential between inlet and exit, lb/ft^2

Positive value indicates outlet pressure greater than inlet.

Negative value indicates outlet pressure less than inlet.

VIII, B, Design of Lines (cont.)

$(Z_2 - Z_1)$ = Change in static head, ft (from 1 to 2)
(+) = Rise in static head
(-) = Loss in static head
H_{losses} = Head losses due to fluid flow, ft
ρ = Fluid density, lb/ft^3

These curves show the pressure difference between the entrance and exit of the line, and do not include the entrance or exit losses themselves.

It will be noted that the plots of pressure drop versus diameter for engines AJ-1, AJ-2, and AJ-3 (Figures VIII-B-1 through -12) include two curves, one labeled "static" or "1-g", the other labeled "10-g acceleration." The 10-g curves illustrate the effect of vehicular acceleration on effective pressure drop in a variety of line applications. The direction of flow (up or down), the vertical distance between line inlet and exit, and the density of the fluid are the primary factors that determine the effect of acceleration on pressure difference from inlet to exit. All pressure drop values plotted in these curves are simply the static pressure at the line inlet less the static pressure at the line exit. This overall pressure drop is obtained from the alegbraic sum of the static head (vertical distance between inlet and exit; positive, if inlet is above exit) and the fluidynamic loss (as previously described from Darcy's equation).

The pressure drops established by this technique form the basis for selecting line diameter when the structural considerations discussed in the following section are not the dictating criteria. In some instances, the maximum allowable pressure drop from an engine system viewpoint is the primary criterion. For most of the lines analyzed in this program, use of the 25- or 50-psi pressure drop allowed by the initial engine-system pressure schedule would result in lines of smaller diameter than those selected. The point on the pressure differential versus diameter curve corresponding to the allowable pressure would be on the steeply sloped portion of the curve, however, and would result in severe changes in pressure drop with minor variations in diameter tolerance. For most of these lines, a diameter corresponding to the knee of the curve is more desirable as a compromise between pressure drop, line flexibility, and sensitivity of pressure drop to line diameter variations.

For hot-gas lines where all the boundary conditions such as fluid-temperature variations, manufacturing tolerances, line-end mismatch, and internal interferences (i.e., bellows restraints) are not known precisely, a flow-velocity limit of Mach 0.3 should be used to ensure that sonic velocity will not be reached, which would choke the flow. For example, the hydrogen-rich hot-gas line of the AJ-8 modular engine (Figure VIII-B-22) will tolerate a temperature increase of 20% at a nominal velocity of Mach 0.3 before the flow reaches sonic velocity, as opposed to only a 10% temperature increase at Mach 0.7.

2. Line-Weight Considerations

As explained in the introduction (Section V), decreasing the weight of auxiliary components is not, in itself, sufficient justification for advanced-technology effort. An interest in the combined effects of high pressure and large diameter on line weight resulted in the specific analysis of three AJ-1 lines (Figures VIII-B-23 through -25) and a parametric analysis of wall thickness and line weight for both high- and low-pressure lines, based on Inconel 718 material (Figures VIII-B-26 through -29). These curves are based on the classical hoop-stress formula:

$$t = \frac{P\,D}{2\,S_a}$$

where: t = Wall thickness, in.

P = Internal pressure, psi

D = Mean diameter, in.

S_a = Allowable stress (108,000 psi was used for Inconel 718 at -200°F)

VIII, B, Design of Lines (cont.)

The effects of line geometry and materials of construction on the overall weight of the line are discussed in the following sections.

3. Line-Structural Considerations and Flexibility

Parametric evaluation of the changes in proportion of the overall dimensions of engine major components, such as turbopumps and thrust chambers, shows that lines may be expected to have lower length-to-diameter ratios for large-thrust, high-pressure engines. While this is to be desired from the pressure-drop viewpoint, the short, large-diameter line presents appreciably more difficult structural-design problems.

The two primary structural considerations for rocket-engine ducting are the end reactions imposed on the mating components and the structural integrity of the line proper. The following discussion covers both the problem of analyzing the line to establish the flexibility and strength requirements and the means of acquiring line flexibility.

a. Line-System Analysis

The end-reaction loads and associated stresses in any given line are the result of the following:

(1) Thermal expansion or contraction of the lines and other hardware connecting the line anchor (terminal) points.

(2) Other forces tending to move the anchor point, such as thrust, vehicle acceleration, or vibration.

(3) Fluid pressure within the line.

(4) Fluid momentum, or dynamic loads from fluid flowing in the line.

(5) Vehicle acceleration.

VIII, B, Design of Lines (cont.)

The approach to the ducting-system-analysis study in this program consists of a detailed evaluation of one representative line from the AJ-1 engine and an investigation of the various means of analysis available to the engine designer. The sample line chosen for analysis was the AJ-1 fuel-pump discharge line to the secondary-combustor cooling jacket, a relatively short line with a single 128° bend. The results of this analysis are shown in Table VIII-B-1 and Figure VIII-B-30. The conditions for which these loads were calculated are based on the engine operating at a pump-discharge pressure of 5500 psi and at the off-design acceleration of 10-g (see Section VI,B,2,b) immediately before burnout. The steady-state operating condition presented the most severe thermal-expansion loads for this particular line; therefore, only these loads are presented. For other applications, additional calculations are required to ascertain if the chill-down or transient thermal conditions might present more severe loads than during steady-state operation.

Table VIII-B-1 and Figure VIII-B-30 show that the end loads resulting from the 10-g acceleration of the vehicle are quite small; not over 0.25% of the total force in any instance. It is also of interest to note that an increase in line diameter results in an increase in reaction load forces from all loads except that due to fluid momentum, which decreases. This may reasonably be expected, because the fluid velocity is reduced when the weight flow remains constant and the diameter is increased.

The total end-reaction forces in this example increase with increasing diameter in each instance. However, in most cases, the stress corresponding to these forces does not change significantly.

VIII, B, Design of Lines (cont.)

The end reactions resulting from thermal expansion and the deflection of major components under operating loads were calculated using the method of piping-system analysis presented in the Marks Handbook* and later verified using the elastic-center method of analysis. The end reactions resulting from acceleration and fluid momentum were determined by the elastic-energy methods of analysis.

In the design of an actual high-pressure engine, a variety of line configurations would normally be analyzed to select the best design for each particular application. An investigation was made into the possibility of adapting presently-available computer techniques for the analyses of piping systems to rocket engine applications. For a long time the presently available procedures have had widespread application in those fields using large piping systems, such as shipboard and stationary steam power plants, petroleum refineries, and chemical processing plants.

A number of firms were found to have the facilities and procedures for the computer analysis of conventional piping systems. Among them are the M. W. Kellogg Company, New York; Service Bureau Corporation (SBC) in Palo Alto, California; Electric Boat Company, Groton, Connecticut; and Aerojet-General's Aetron Division in Covina, California. The majority of these computer programs are based on a 6 by 6 matrix system and are capable of directly handling pressure load, thermal load, physical movement of end points, and concentrated loads. Acceleration loads, line stiffening due to high pressure, distributed dynamic loads around bends (fluid momentum), effects of vibration, transient loads, and water hammer are considered with some programs. Acceleration, fluid momentum, and other distributed loads must be considered as a series of concentrated loads. SBC offers a 24-hr specialized computer service for piping-system analysis.

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*L. S. Marks, Mechanical Engineers' Handbook, (New York, Inc., McGraw-Hill, 5th Edition, 1951) p. 478.

M. W. Kellogg Co., Design of Piping Systems, 2nd Ed., Wiley, New York (1956).

VIII, B, Design of Lines (cont.)

To evaluate the feasibility and desirability of developing a similar program particularly tailored to the requirements of high-pressure rocket engines, Dr. John E. Brock, Professor of Mechanical Engineering, U. S. Naval Post Graduate School, Monterey, California, was consulted. The optimum program was considered to be one into which the design engineer could input the parameters of line configuration, end movement, line-material properties, fluid properties, flow rate, etc., and receive back the calculated end-reactions and line stresses. On investigation, it was determined that the expense involved in establishing a flexible program sufficiently sophisticated to handle such problems would outweigh the potential gains. Instead, it is recommended that the analysis of rocket-engine ducting systems make use of the programs presently available for conventional piping systems.

Such an approach would consist of performing, by hand, the relatively simple calculations necessary to approximate the distributed loads resulting from fluid momentum and vehicle acceleration as concentrated loads. These concentrated loads could then be used, along with the end-motion and thermal-expansion data obtained from existing programs, such as those mentioned above. In this fashion, a number of candidate lines of different diameter and configuration could be rapidly and inexpensively analyzed for each application.

b. Means of Acquiring Line Flexibility

There are three methods of obtaining greater flexibility in a line or duct to minimize end-reaction forces or stresses within the line: (1) alter line geometry, (2) alter line material properties, or (3) add flexible joints to the line.

VIII, B, Design of Lines (cont.)

(1) Line Geometry

Variations of line geometry usually take the form of changing line diameter, line configuration, or both. Small-diameter lines are more flexible than large-diameter lines, but reduction in diameter is usually limited by pressure-drop considerations, as discussed previously. Flexibility may also be increased by changing the line configuration, usually by increasing the total length of the line. Because any increase in length of line may be expected to increase the pressure drop, caution is required. Changes in number and types of bends will also have an effect on pressure drop.

In conventional piping-system analysis, the following expression is normally used to indicate the inherent flexibility of the system to aid in deciding whether or not detailed analysis of the particular section is necessary.

$$\frac{dy}{(L-u)^2} \leq 0.03$$

where: d = Nominal diameter, in.
y = Deflection (direction unspecified), in.
L = Length (of duct center line), ft
u = Anchor to anchor distance, ft

If the above is true, the line is considered flexible and detailed analysis need not be performed. Obviously such a criterion is very approximate, because it does not consider the direction of deflection, gimbaling or orientation of anchor points, material properties, or even wall thickness. However, the expression is useful for roughly evaluating the relative flexibility of alternative configurations.

VIII, B, Design of Lines (cont.)

For example, consider the two alternative arrangements of the AJ-8 oxidizer-rich hot-gas line shown in Figures VIII-B-20 and -21. The small differences in pressure-drop characteristics have already been pointed out. The difference in flexibility is significant, however. Using the flexibility approximation, these two line configurations were evaluated for a 12-in. diameter with the result that the single 180° bend would be considered flexible up to a deflection (y) of 0.046 in., as compared with only 0.020 in. for the version with two 90° elblows.

This same expression also provides a convenient illustration of the severity of a particular rocket-engine line problem, such as the sample AJ-1 fuel-pump discharge line (Figure VIII-B-2) that was analyzed in detail. Considering nominal line diameter of 5-in., the flexibility approximation yields a maximum acceptable deflection of 0.012 in. The actual deflection required for this line due to thermal expansion alone is approximately 0.6 in. horizontal and 0.4 in. vertical. The detailed analysis of this line showed that line stresses for this particular condition did not exceed 20,000 psi, and the line would be feasible if the mating hardware could tolerate the moderate end-reaction loads.

The Summary of Design Criteria (Table II-1) used an even simpler version of this expression ($\frac{d}{(L-u)^2}$), with all values in inches, to approximate the relative flexibility of the various lines evaluated under this program.

It may be concluded, therefore, that use of such an approximation from conventional piping analysis may prove useful for rough, relative evaluation of candidate line geometries, but comprehensive analysis of the designs is necessary to accurately determine the flexibility of different geometries.

VIII, B, Design of Lines (cont.)

The effect of geometry change on line weight should also be evaluated in the analysis of lines. Again considering the two AJ-8 hot-gas line choices (Figures VIII-B-20 and -21), the 180° bend increases the line length and weight by 18.5%, or roughly 34 lb. This result was obtained by using Figure VIII-B-29 (without correcting unit weight for this higher-temperature application).

In addition to changing the design of a particular line or duct, effective changes in line geometry may frequently be achieved by modifying the location or flange orientation of the parts in mating components, such as pumps and valves.

(2) Line-Material Properties

Another means of obtaining flexibility in a line or duct is to reduce the elastic modulus of the wall material and/or increase the allowable tensile stress. To verify this, a simple parametric analysis was performed on a duct loaded like a simple cantilever beam (Appendix G). Results of this analysis showed that

$$f \sim \left(\frac{S}{E}\right)$$

where:

- f = Flexibility, or tip deflection of cantilever per unit length (dimensionless)
- S = Allowable tensile stress, psi (in the hoop, or circumferential direction)
- E = Young's modulus, or modulus of elasticity, psi (in the axial direction)

VIII, B, Design of Lines (cont.)

and that

$$PF \sim \frac{S^2}{\rho E}$$

where: PF = Performance factor, or flexibility per unit weight

ρ = Wall material density, lb/in.^{3}

These relationships were used to evaluate the potential advantages of high-strength, low-modulus, filament-wound ducting discussed in Section VIII, F.

Another application of this information is to the general flexibility criteria discussed in line geometry, above. The expression

$$\frac{dy}{(L-u)^2} \leq 0.03$$

is used with low strength, high modulus materials found in conventional piping systems. It would therefore appear more realistic to modify the expression for use with the high-performance materials considered in rocket engine application. A suggested approach would be

$$\frac{dy}{(L-u)^2} \leq K_m$$

where K_m = material factor

$$= 0.03 \left[\frac{f(S)}{f(E)}\right]$$

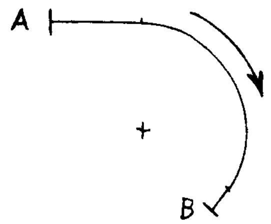
$$= 0.03 \left[\frac{\frac{S}{10x10^3}}{\frac{E}{30x10^6}}\right]$$

VIII, B, Design of Lines (cont.)

the common allowable piping system stress being about 10,000 psi with a corresponding modulus of about 30×10^6 psi. The applicablility of the flexibility expression still remains limited, as explained previously (Section VIII, B,3,b, (1)).

TABLE VIII-B-1

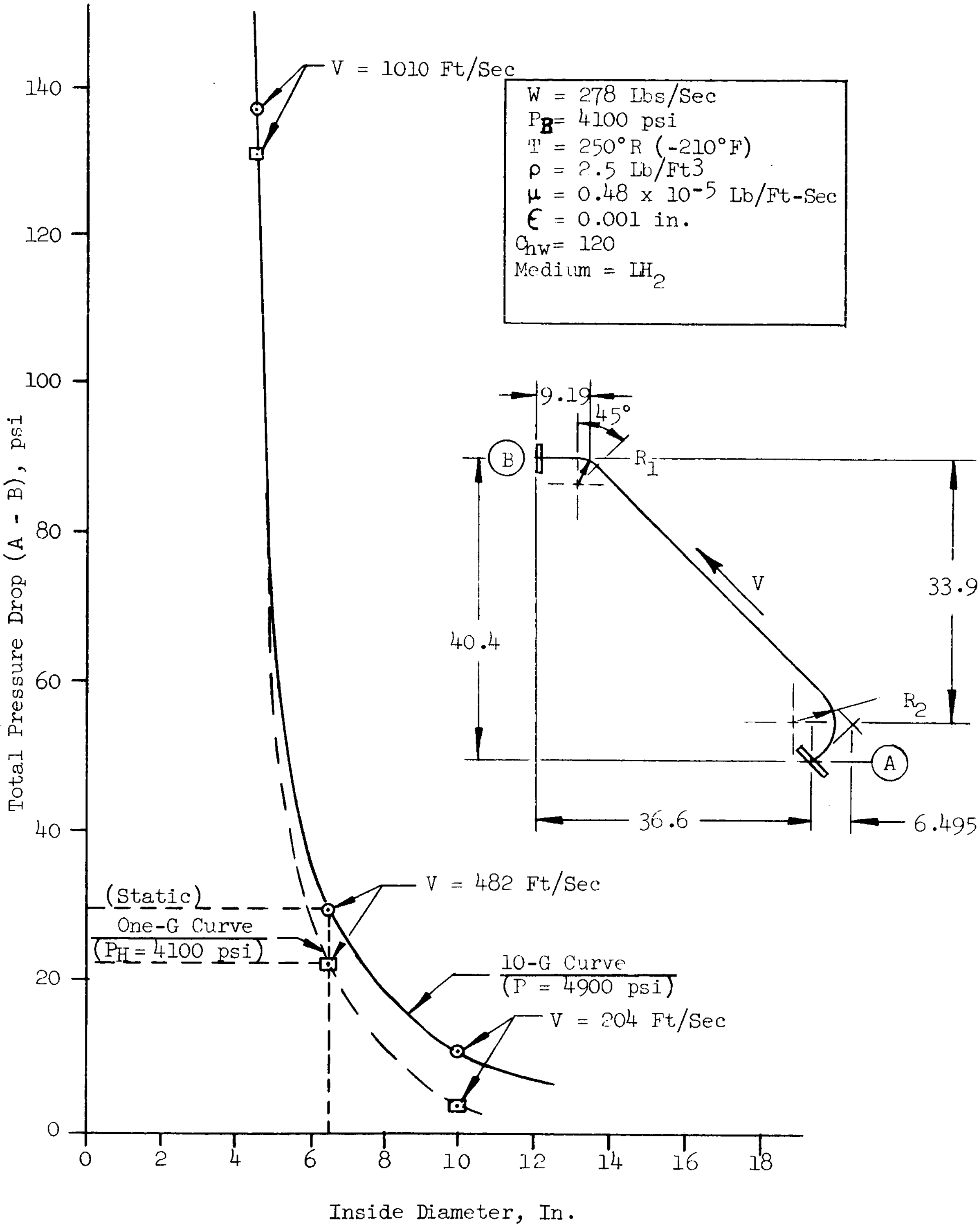
STRUCTURAL ANALYSIS OF AJ-1 FUEL PUMP DISCHARGE LINE TO SECONDARY COMBUSTION COOLING JACKET



w = 278 lb/sec LH_2

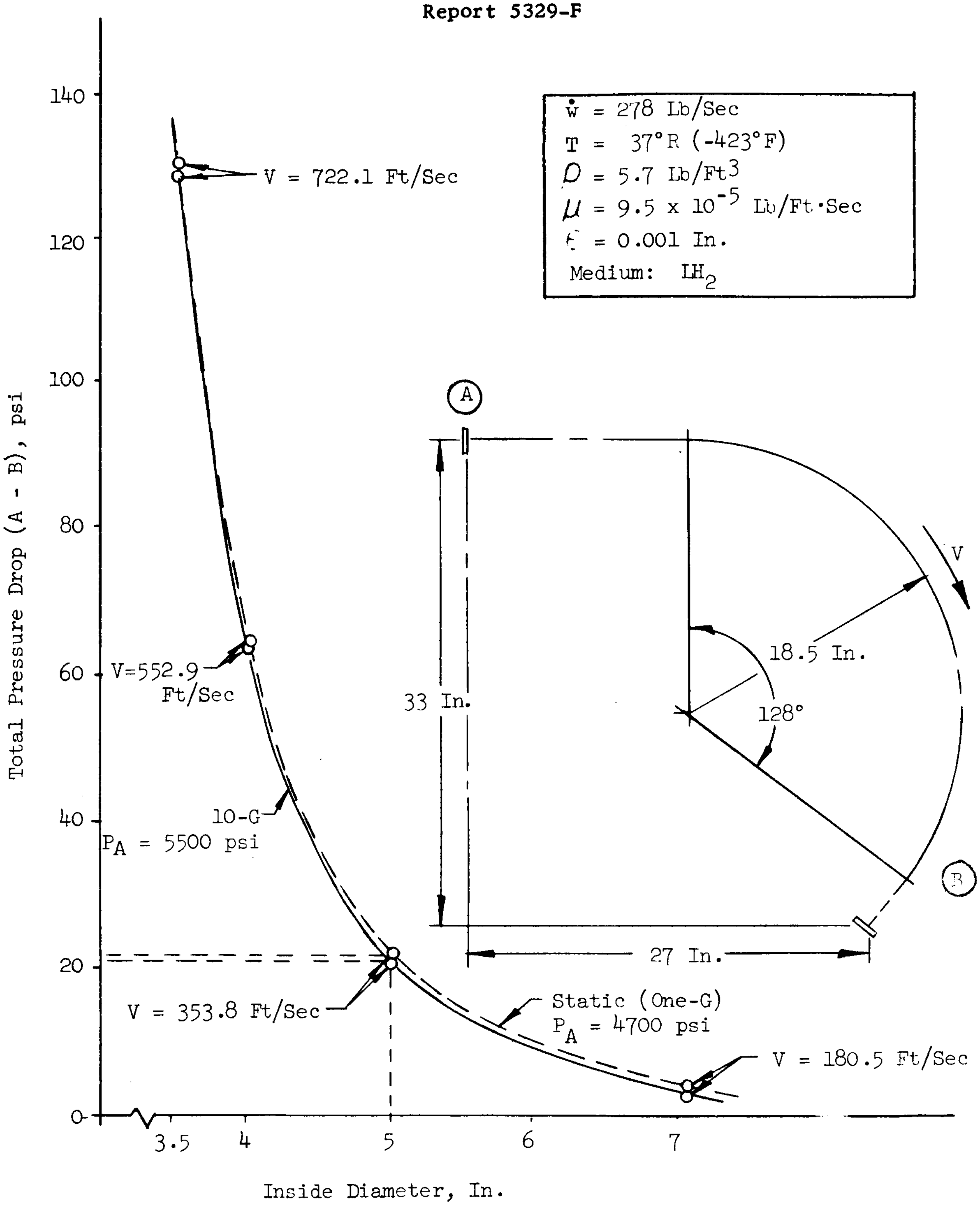
LOAD	SUMMARY OF REACTION FORCES								
	END A								
	+ F_N (LBS)			+ F_S (LBS)			+ M_o (IN-LBS)		
	3.5	5	7	3.5	5	7	3.5	5	7
Thermal Expansion & Engine Deflection	150.6	6400	17,170	1894	8000	26,300	37,837	170,500	578,967
Acceleration (10G Vertical)	0.8	1.1	3.3	29.7	76	137	315	755.4	1386
Momentum Flux ($\dot{w}$ = 278 lb/sec)	941.2	379.5	194.5	5808.9	2585.9	1322.0	90,488.0	37,482.1	19,192.5
Internal Pressure (5500 psi)	52,900	108,000	211,800	-	-	-	-	-	-
TOTAL FORCE	55,347	114,777	229,158	7732.6	10661.9	27,759	52,966	132,262	558,388
Corresponding Stress (psi)	60,950	56,400	63,060	1,703	10,500	15,280	±66,600	±52,000	±87,750
	END B								
	+ F_N'			+ F_S'			+ M_o'		
	3.5	5	7	3.5	5	7	3.5	5	7
Thermal Expansion & Engine Deflection	1064	2364	10,155	2353	9,968	29,720	36,397	165,700	435,477
Acceleration (10G Vertical)	106.0	263.0	478	129.0	319.0	584	234	358	970
Momentum Flux ($\dot{w}$ = 278 lb/sec)	4,346	3337.0	1701.0	6484	3029.0	1547.0	92,006	28,525	15,690
Internal Pressure (5500 psi)	52,900	108,000	211,800	-	-	-	-	-	-
TOTAL FORCE	56,076	108,710	202,868	8,708	12,678	30,683	55,843	136,817	418,817
Corresponding Stress (psi)	61,760	53,400	55,925	19,180	12,450	16,890	±70,220	±53,600	±65,820

Table VIII-B-1



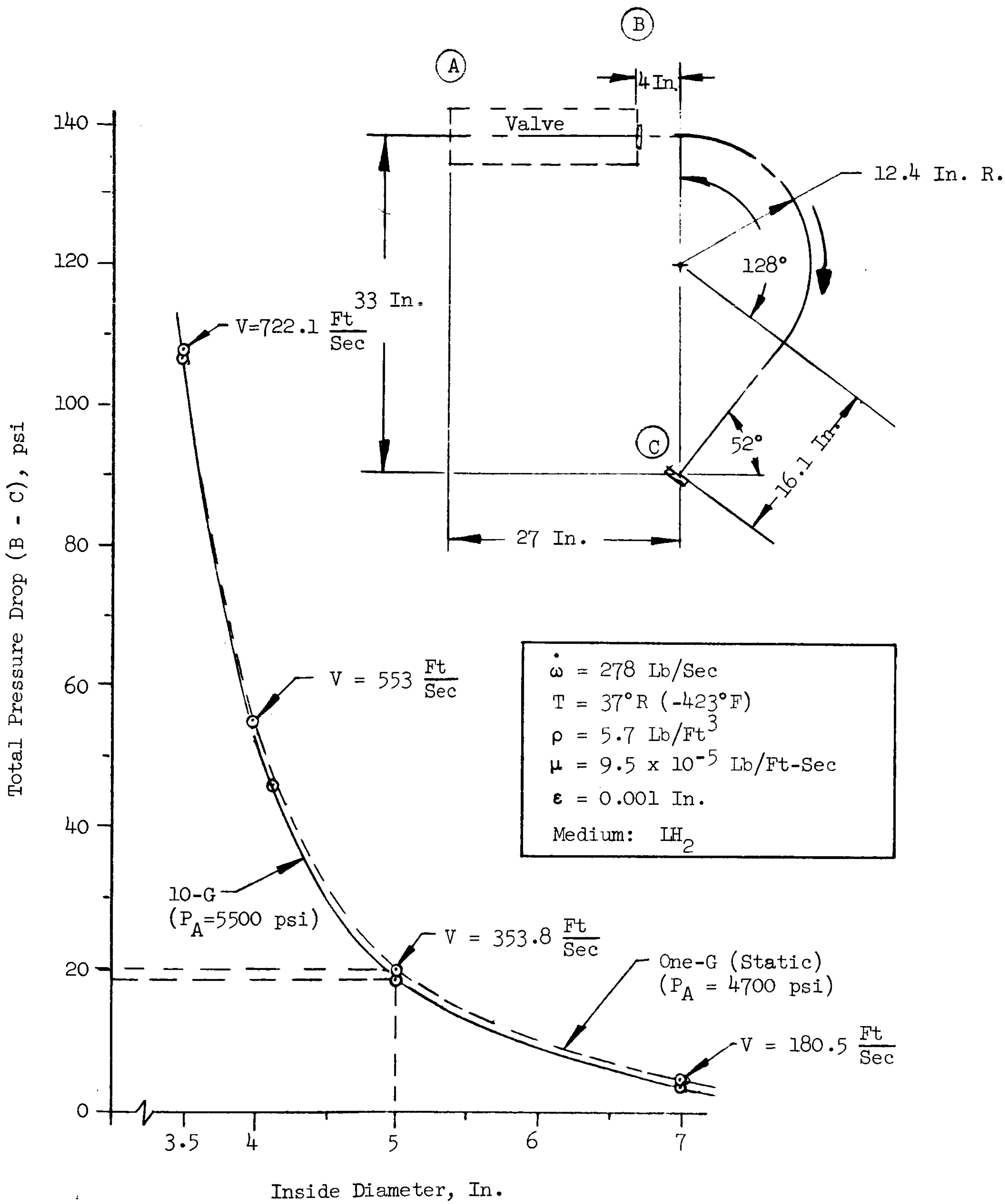
Pressure Drop vs Inside Diameter, Fuel Line to Primary Combustor for the AJ-1 Engine

Figure VIII-B-1



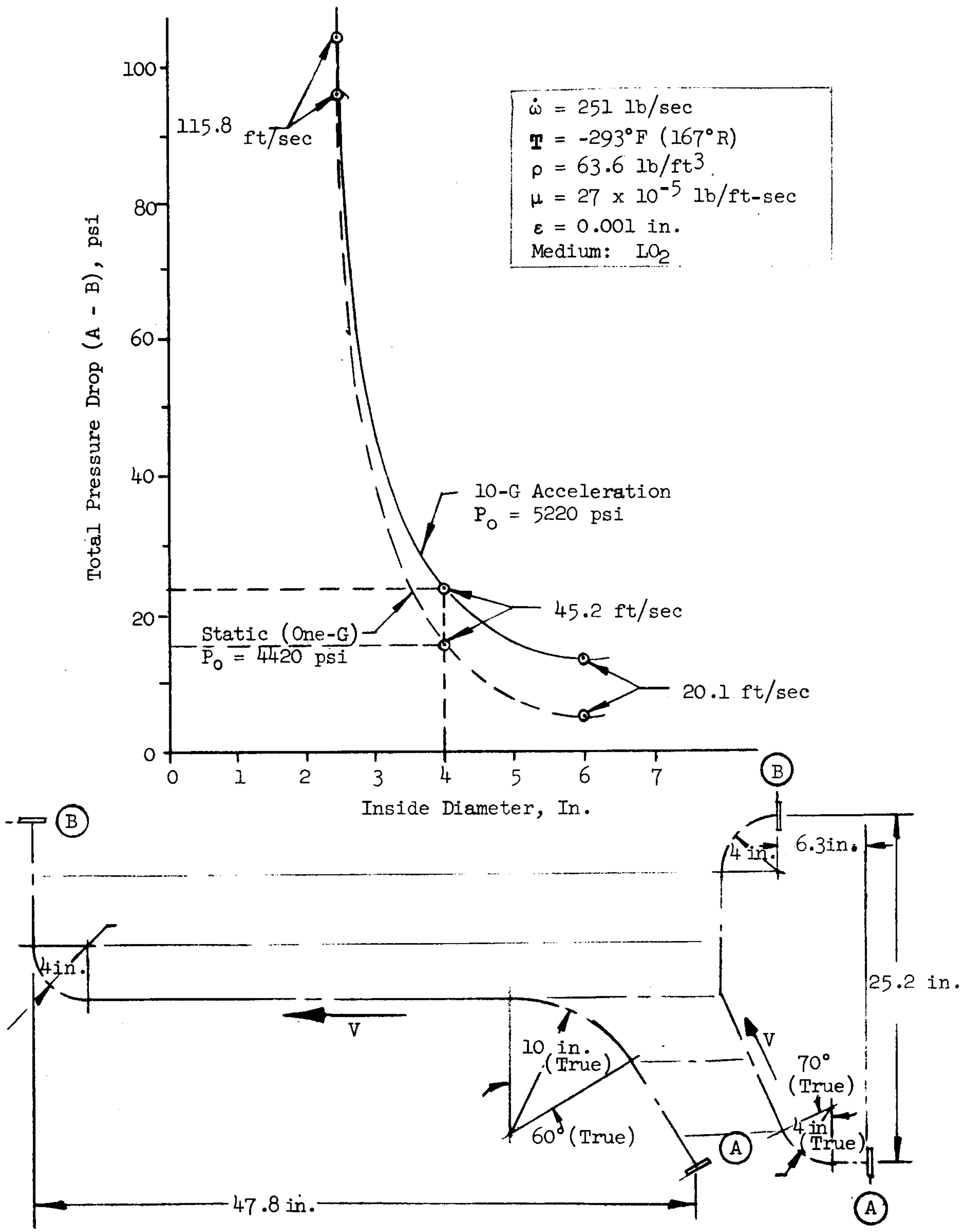
Pressure Drop vs Inside Diameter, Fuel Line to Secondary Combustor Cooling Jacket for the AJ-1 Engine

Figure VIII-B-2



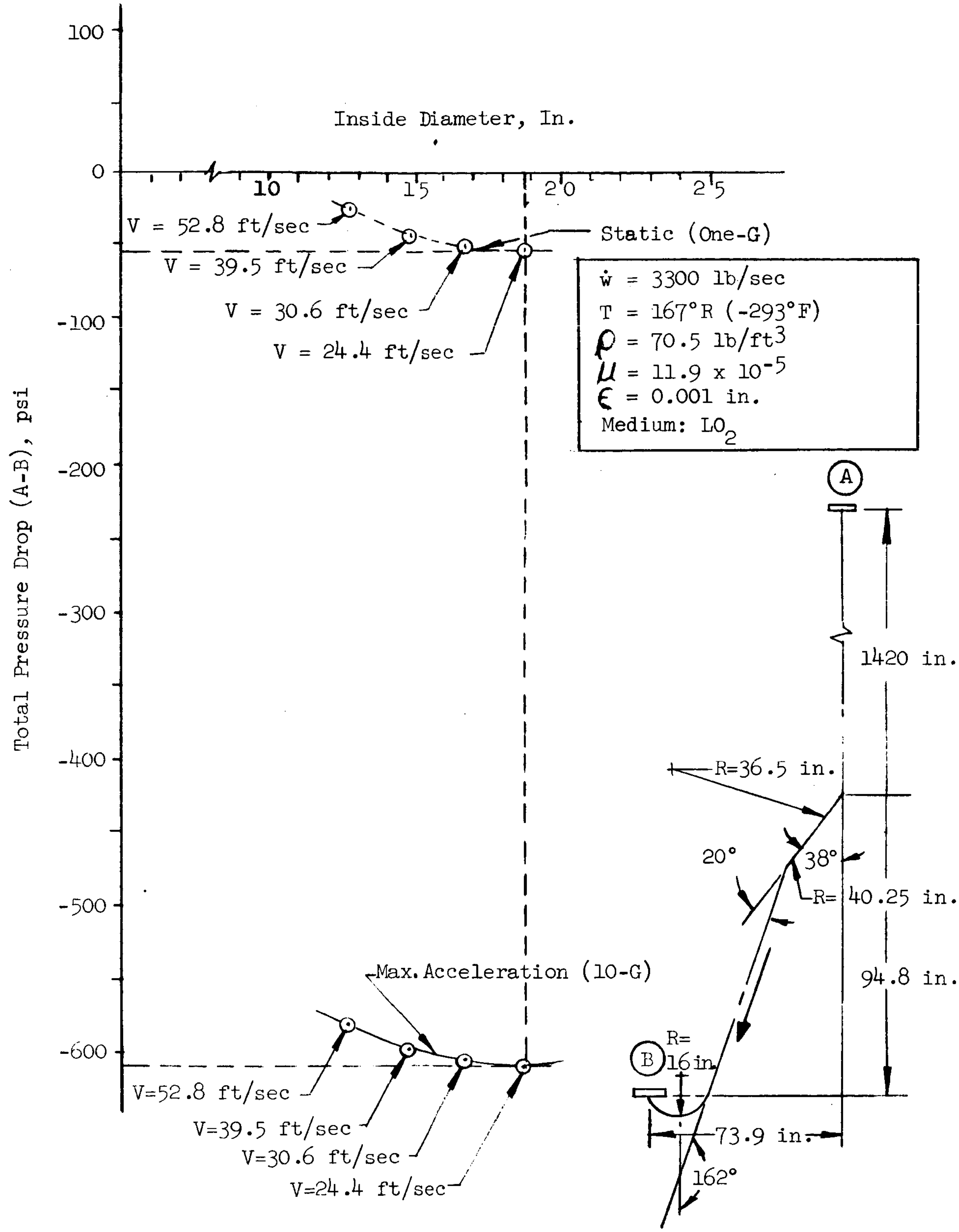
Pressure Drop vs Inside Diameter, Fuel Line to Secondary Combustor Cooling Jacket for the AJ-1 Engine (with Valve)

Figure VIII-B-3



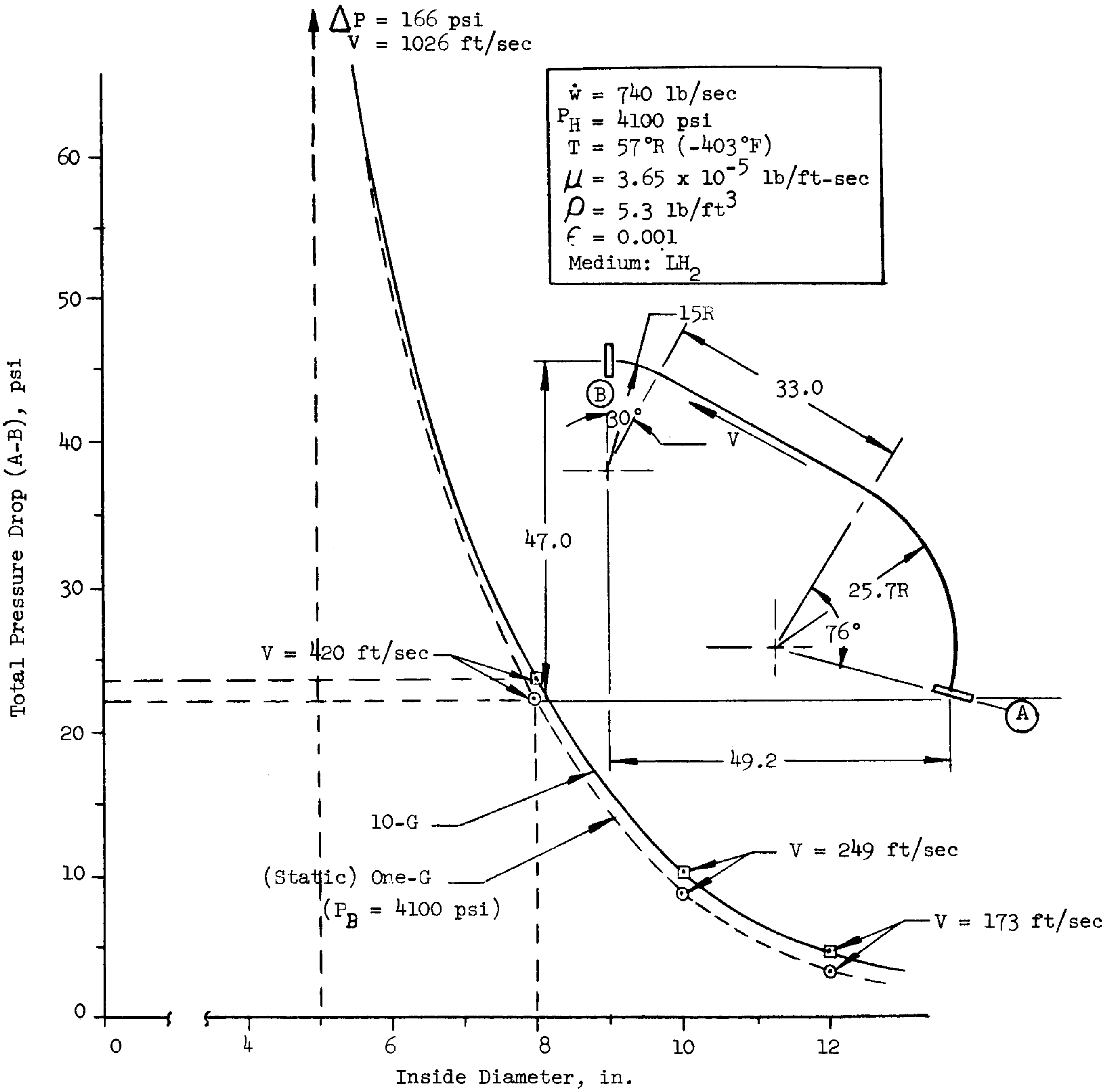
Pressure Drop vs Inside Diameter, Oxidizer Line to Primary Combustor for the AJ-1 Engine

Figure VIII-B-4



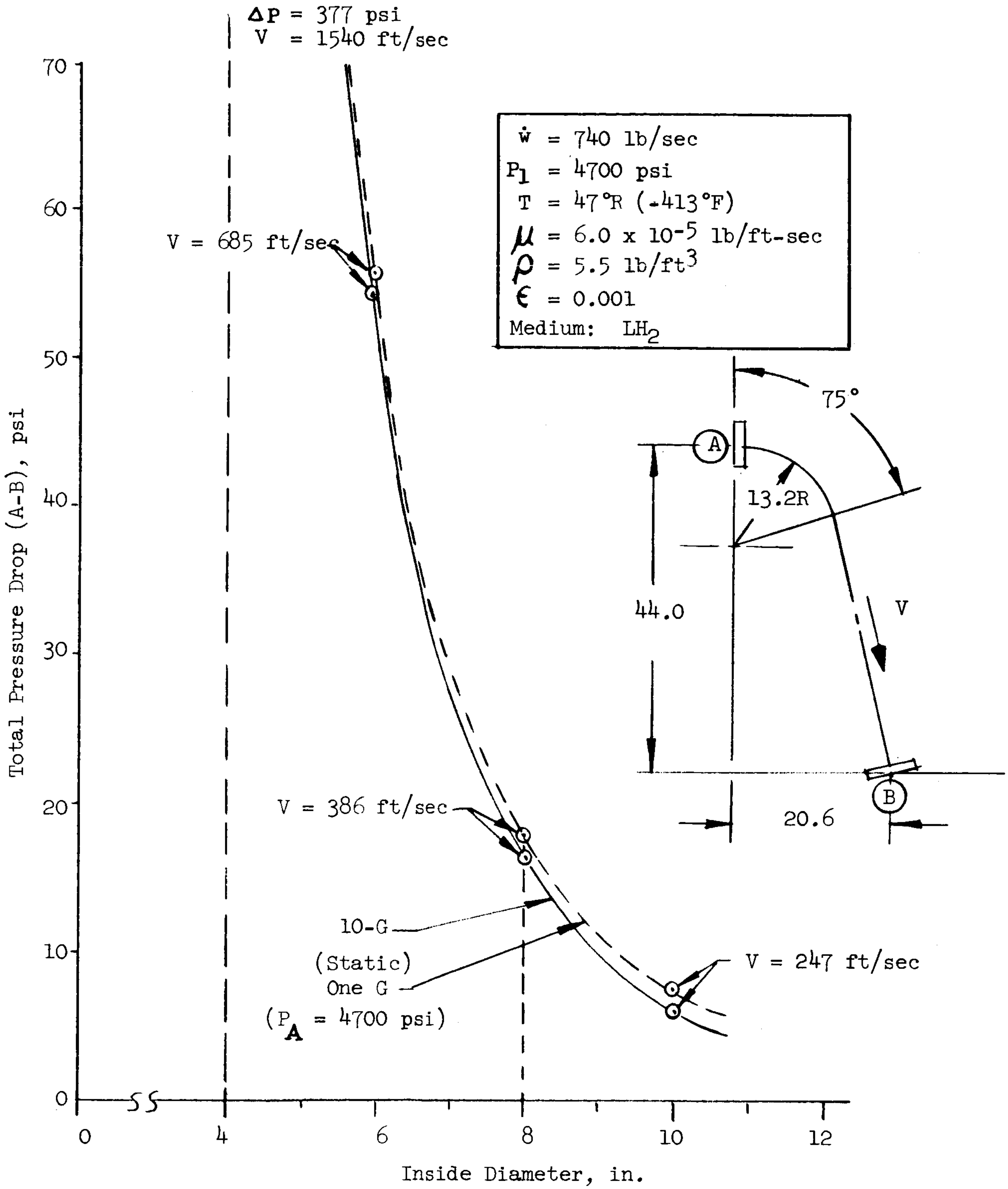
Pressure Drop vs Inside Diameter, Oxidizer Suction Line for the AJ-1 Engine

Figure VIII-B-5



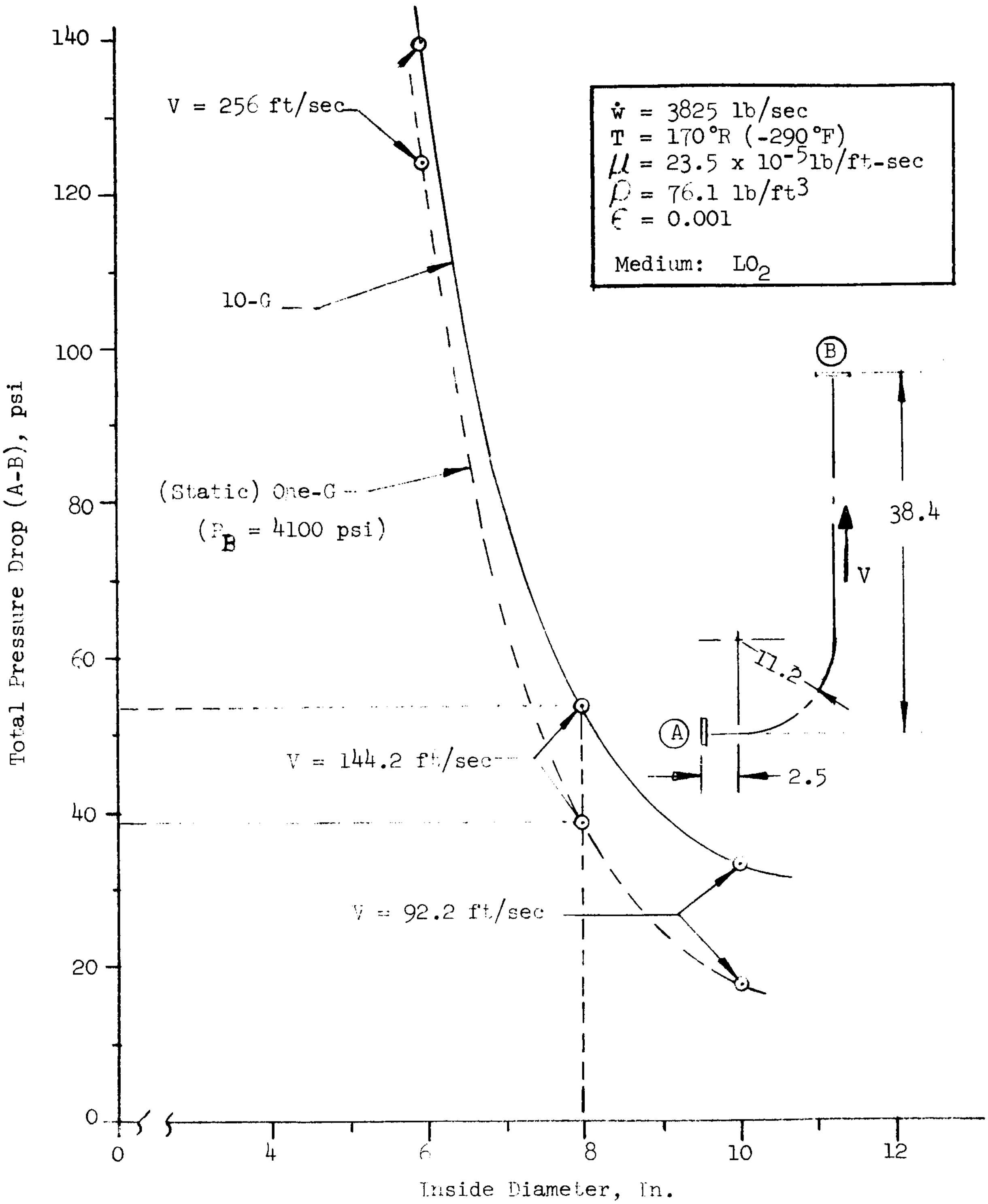
Pressure Drop vs Inside Diameter, Fuel Line to Primary Combustor for the AJ-2 Engine

Figure VIII-B-6



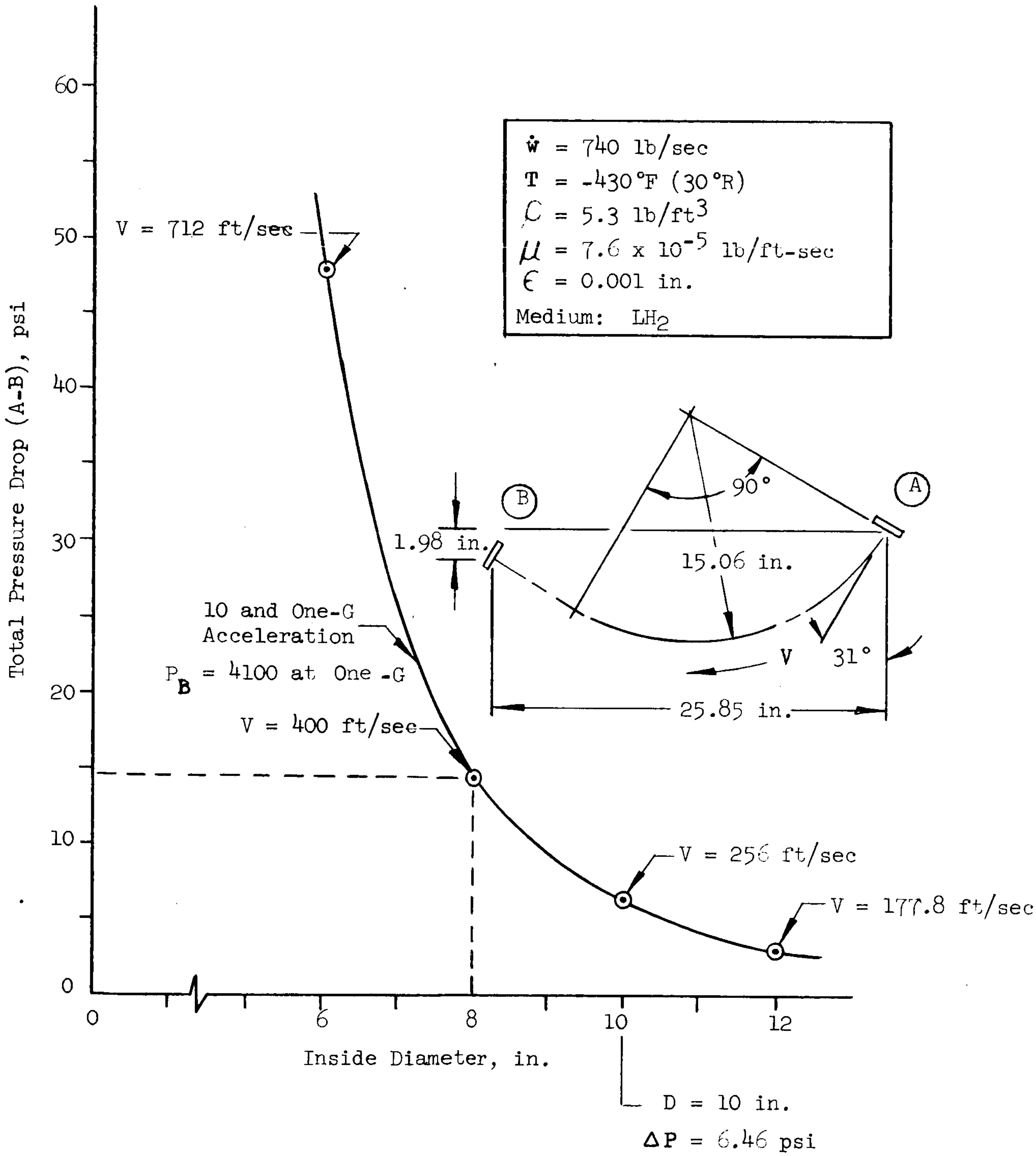
Pressure Drop vs Inside Diameter, Fuel Line to Secondary Combustor for the AJ-2 Engine

Figure VIII-B-7



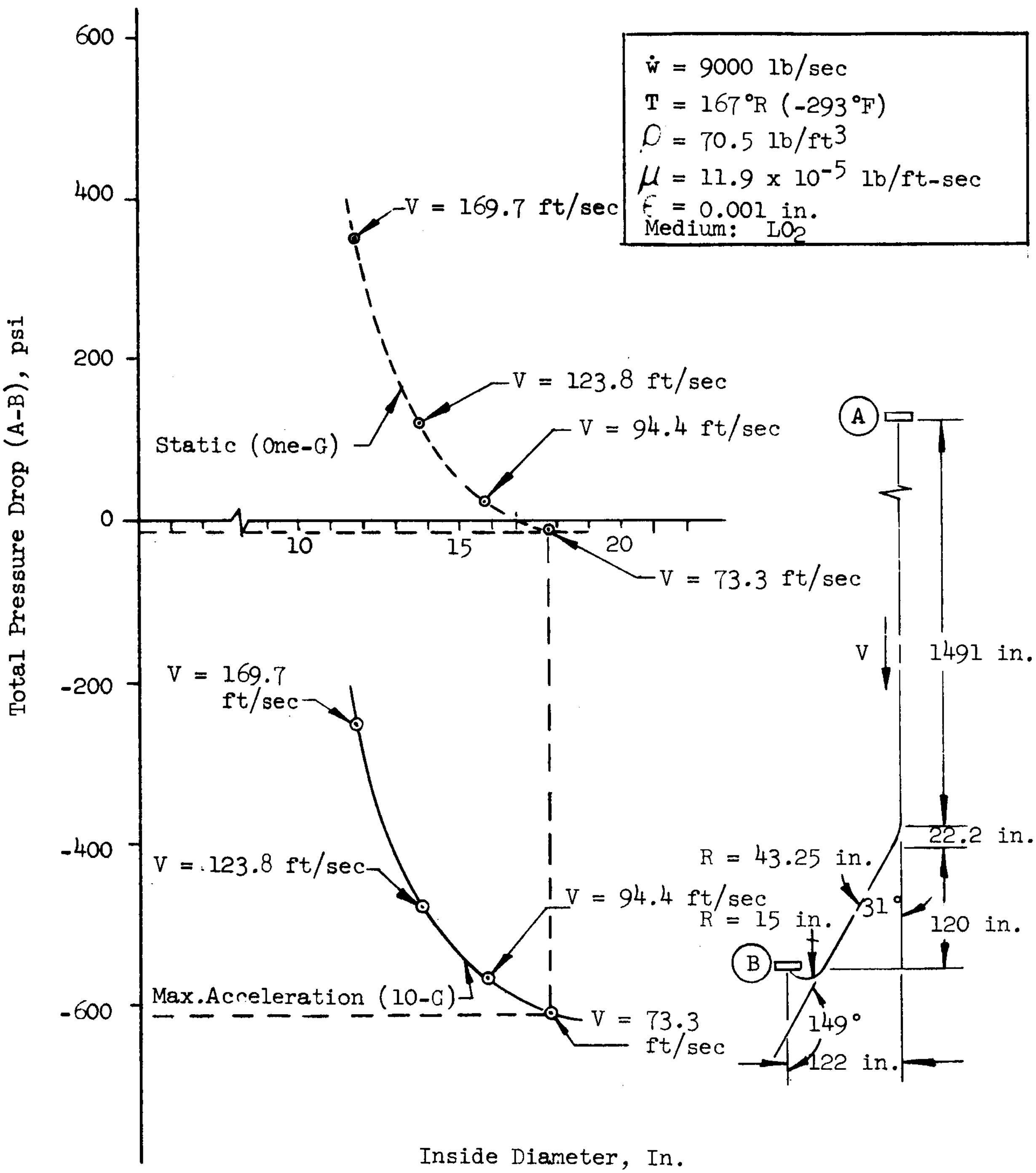
Pressure Drop vs Inside Diameter, Oxidizer Line to Secondary Combustor for the AJ-2 Engine

Figure VIII-B-8



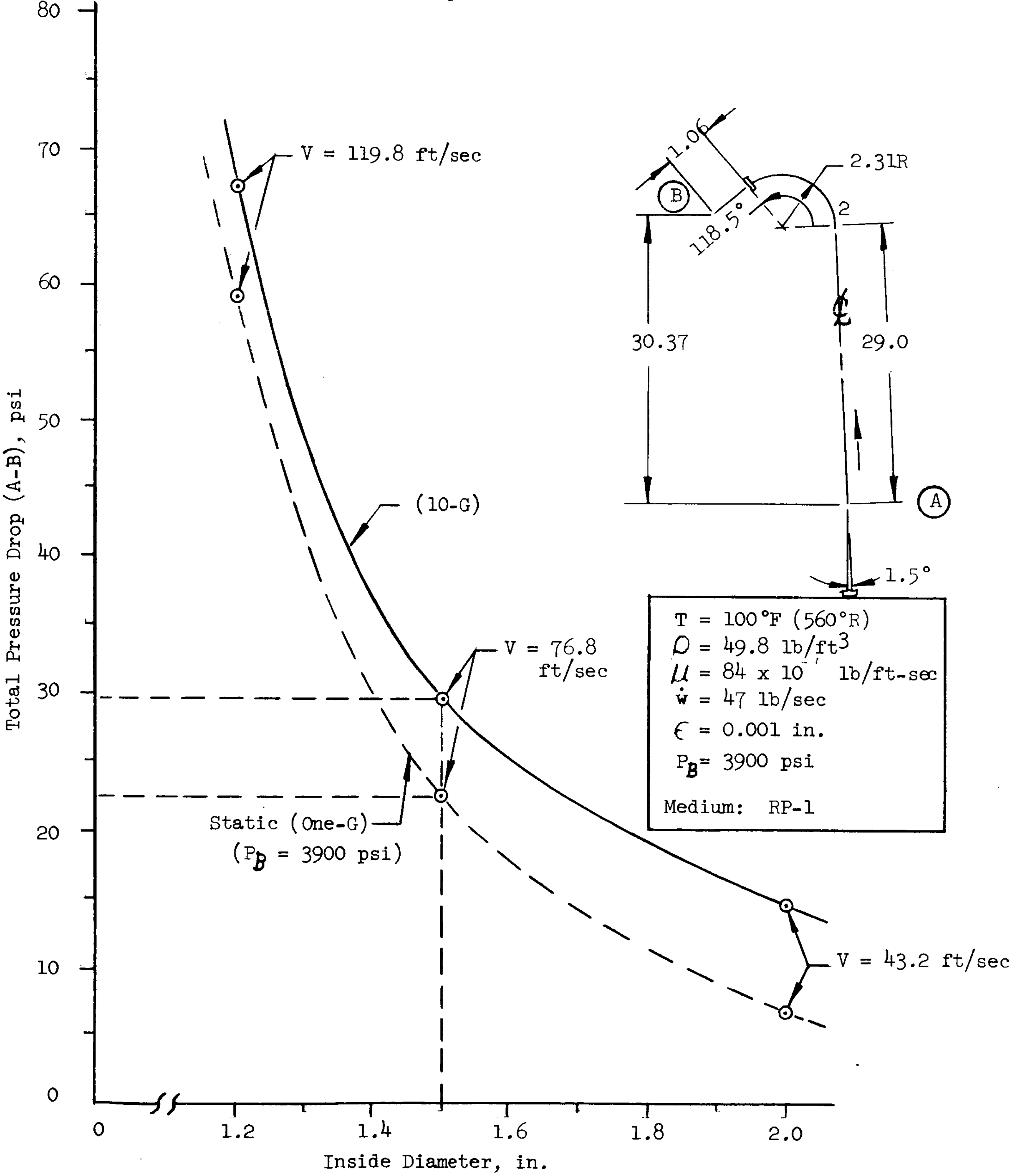
Pressure Drop vs Inside Diameter, Fuel Line to Primary Combustor for the AJ-2 Engine

Figure VIII-B-9



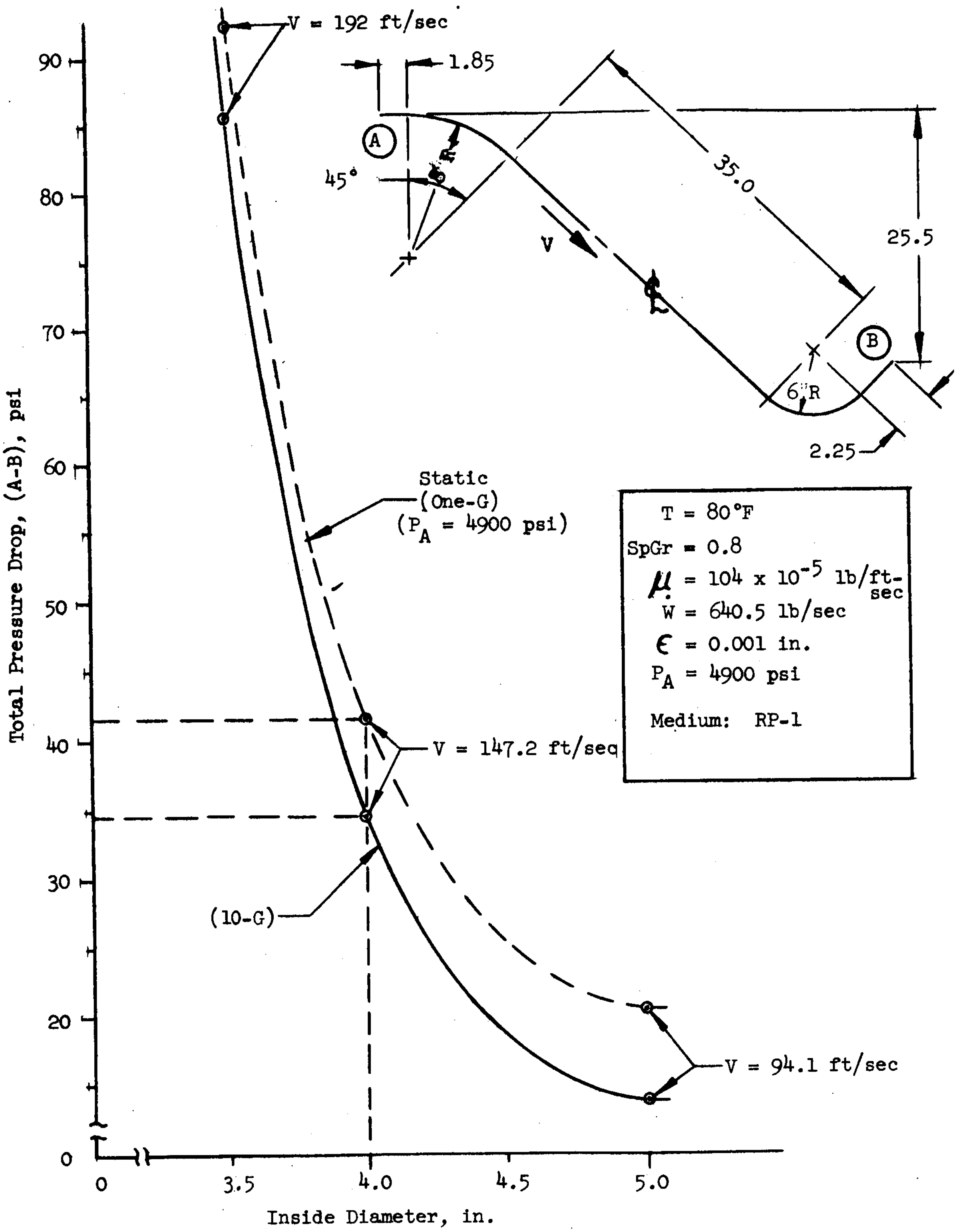
Pressure Drop vs Inside Diameter, Oxidizer Suction Line for the AJ-2 Engine

Figure VIII-B-10



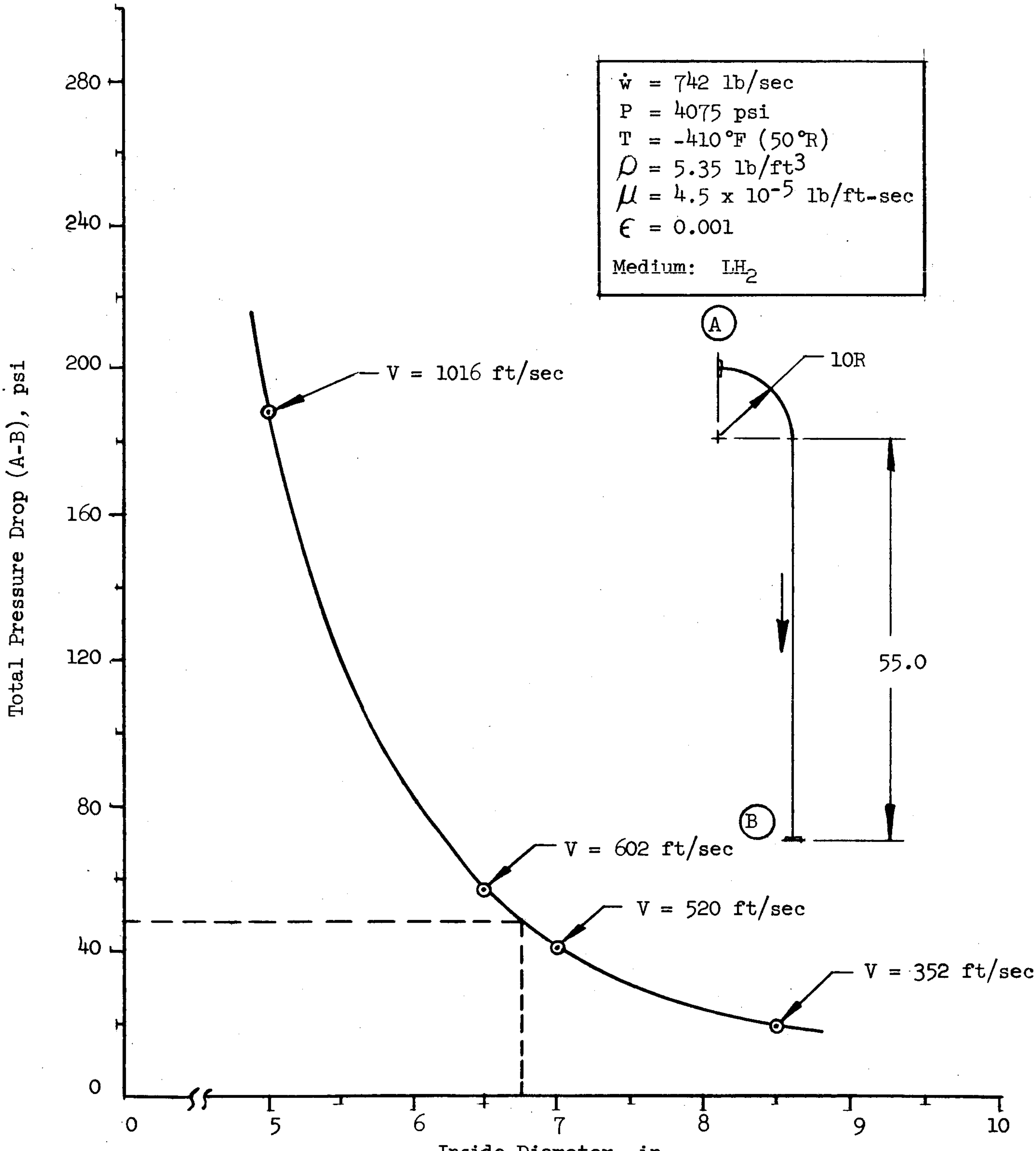
Pressure Drop vs Inside Diameter, Fuel Line to Primary Combustor for the AJ-3 Engine

Figure VIII-B-11



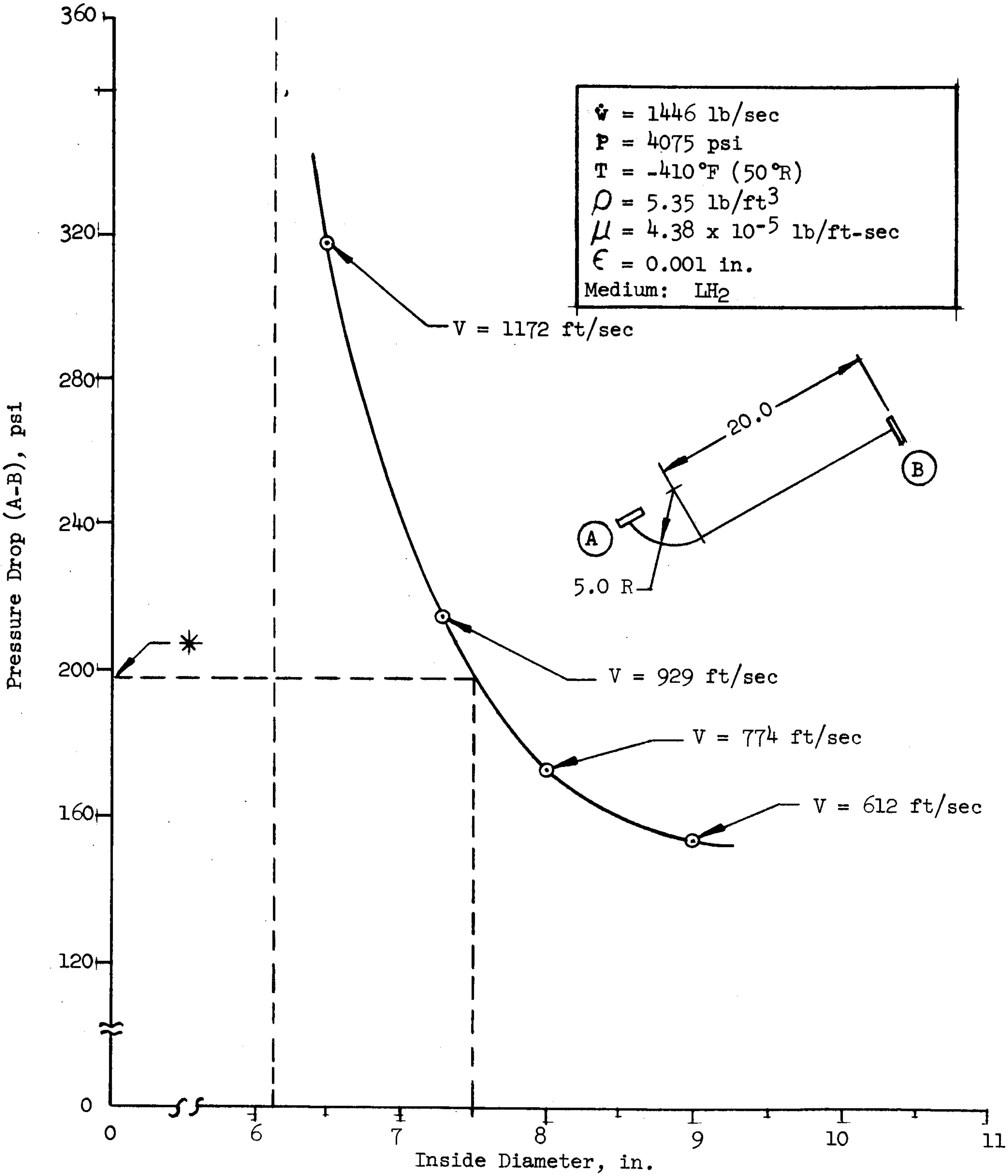
Pressure Drop vs Inside Diameter, Fuel (RP-1) Discharge Line for the AJ-3 Engine

Figure VIII-B-12



Pressure Drop vs Inside Diameter, Fuel Pump Discharge Line to Combustion Chamber Coolant Manifold, AJ-5 Engine

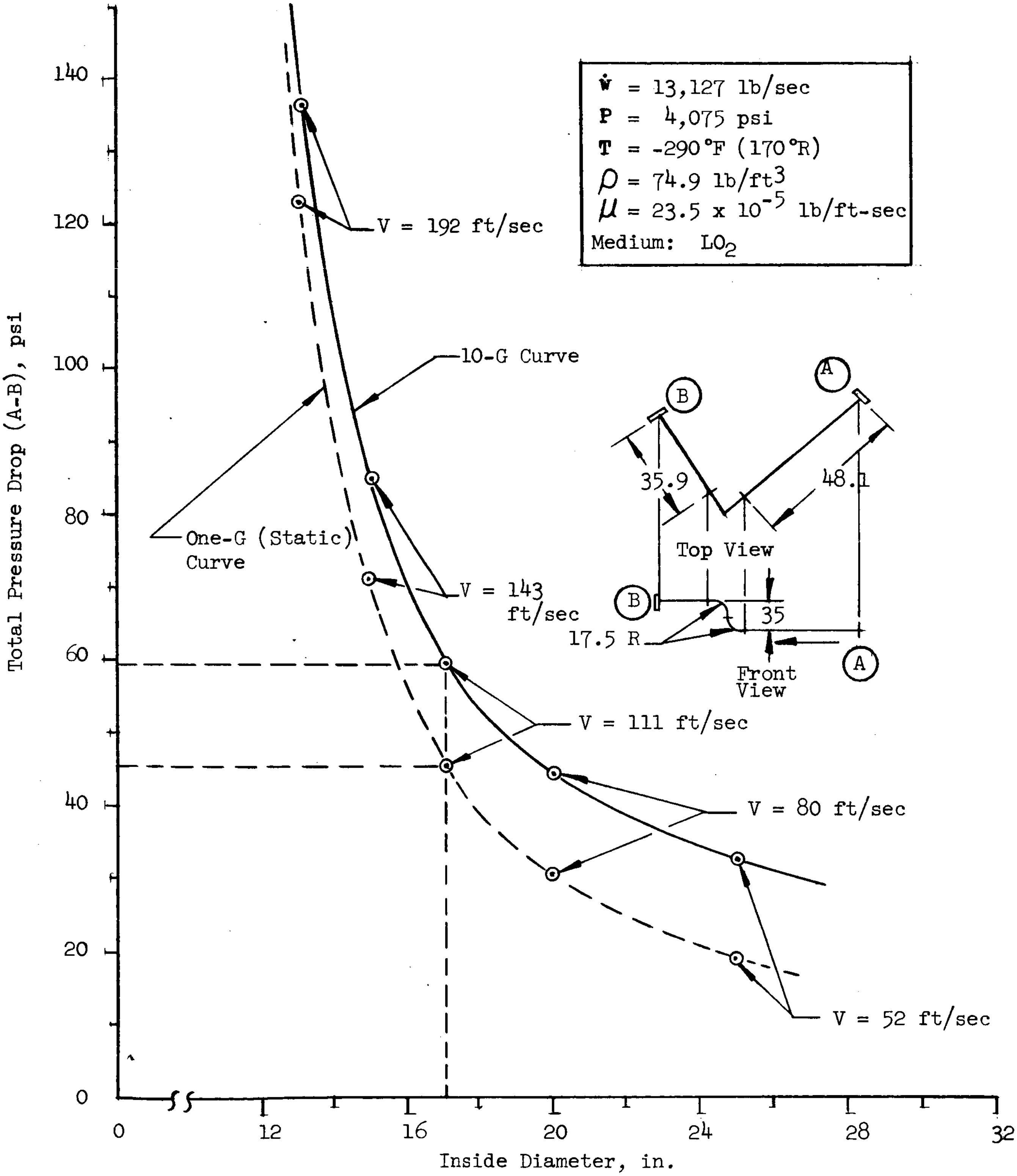
Figure VIII-B-13



* Orifice Plate Required to Obtain 298-psi Pressure Drop to Balance Fuel System.

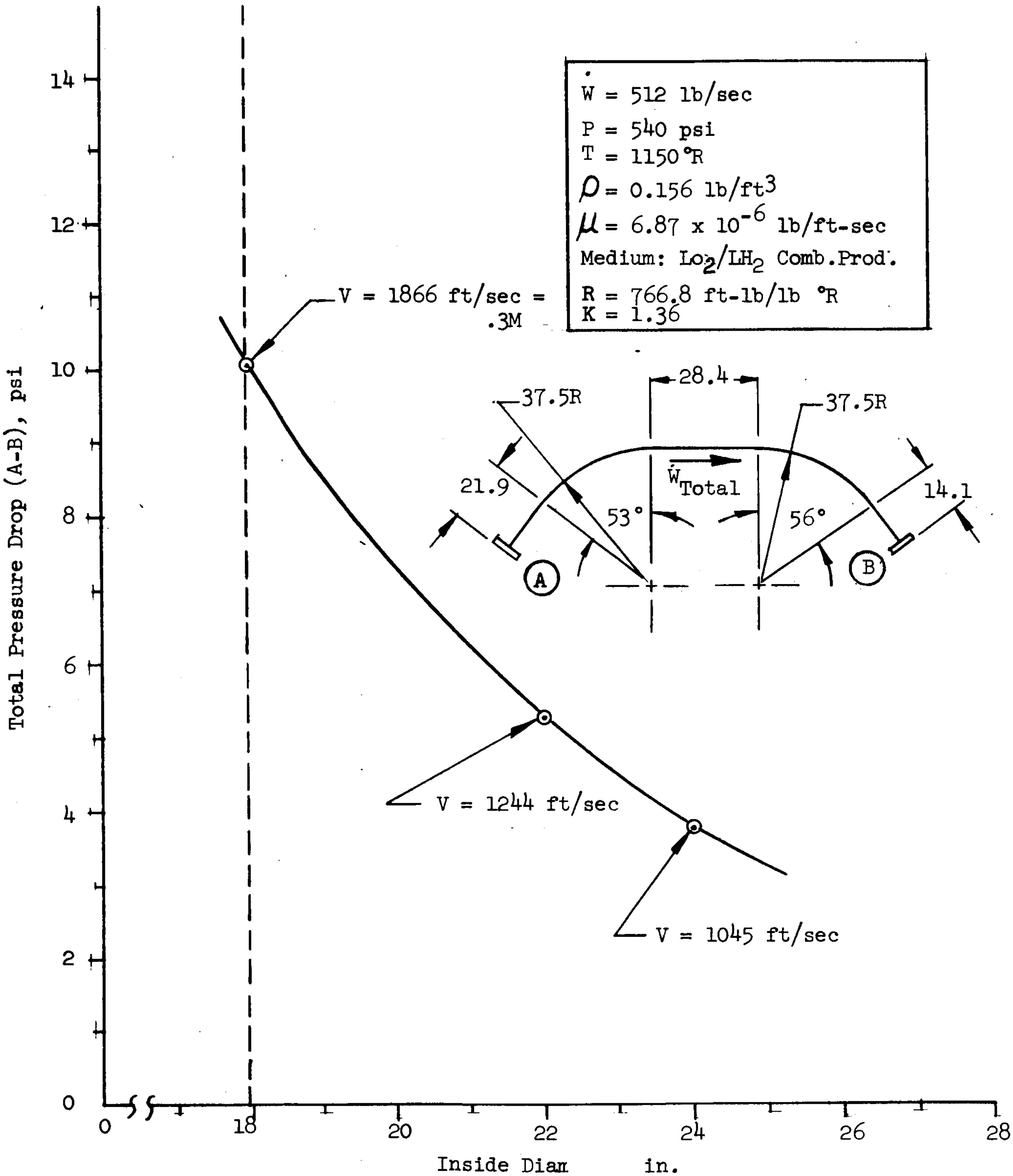
Pressure Drop vs Inside Diameter, Fuel Pump Discharge Line to Injector AJ-5 Engine

Figure VIII-B-14



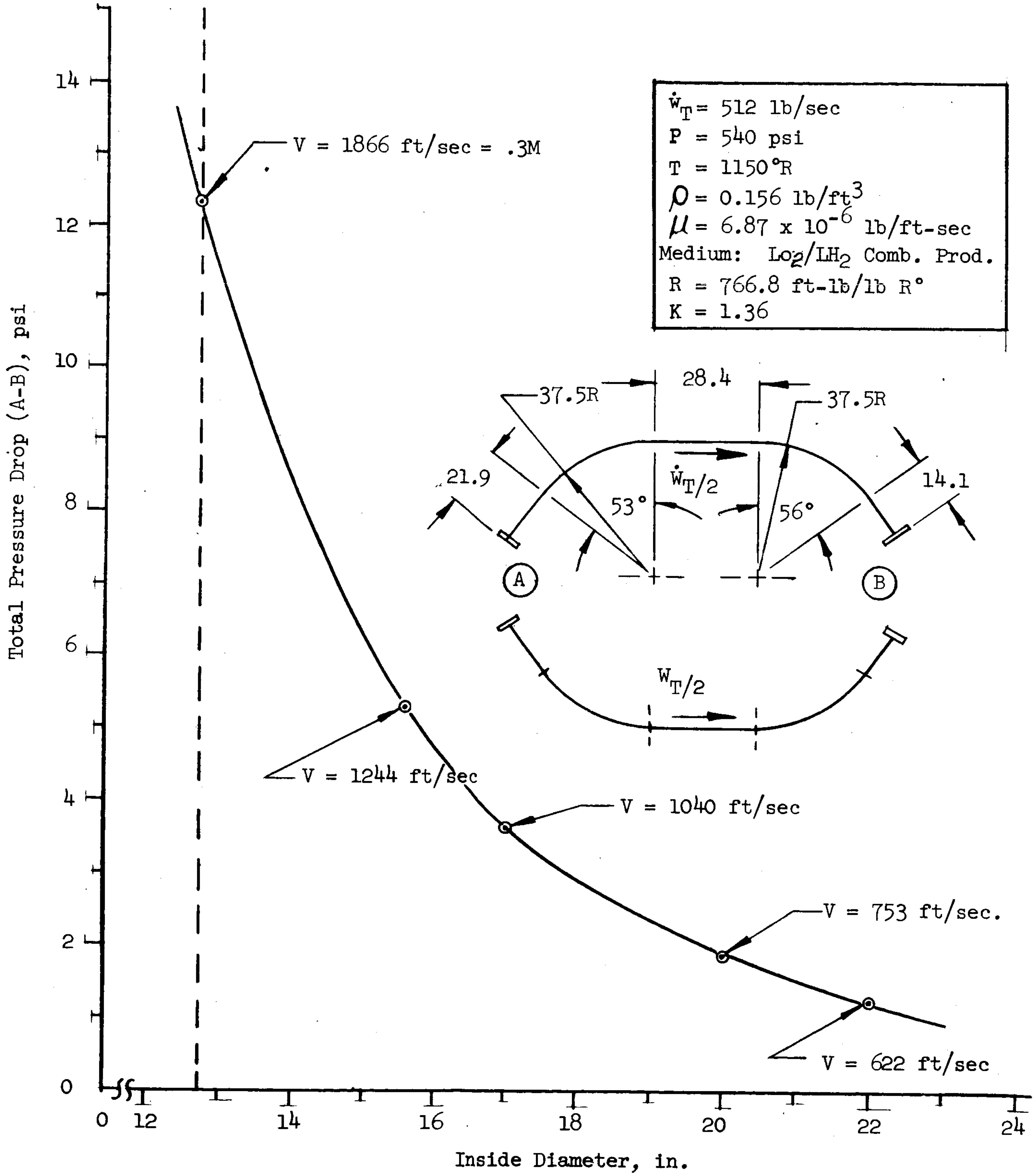
Pressure Drop vs Inside Diameter, Oxidizer Pump Discharge Line, AJ-5 Engine

Figure VIII-B-15



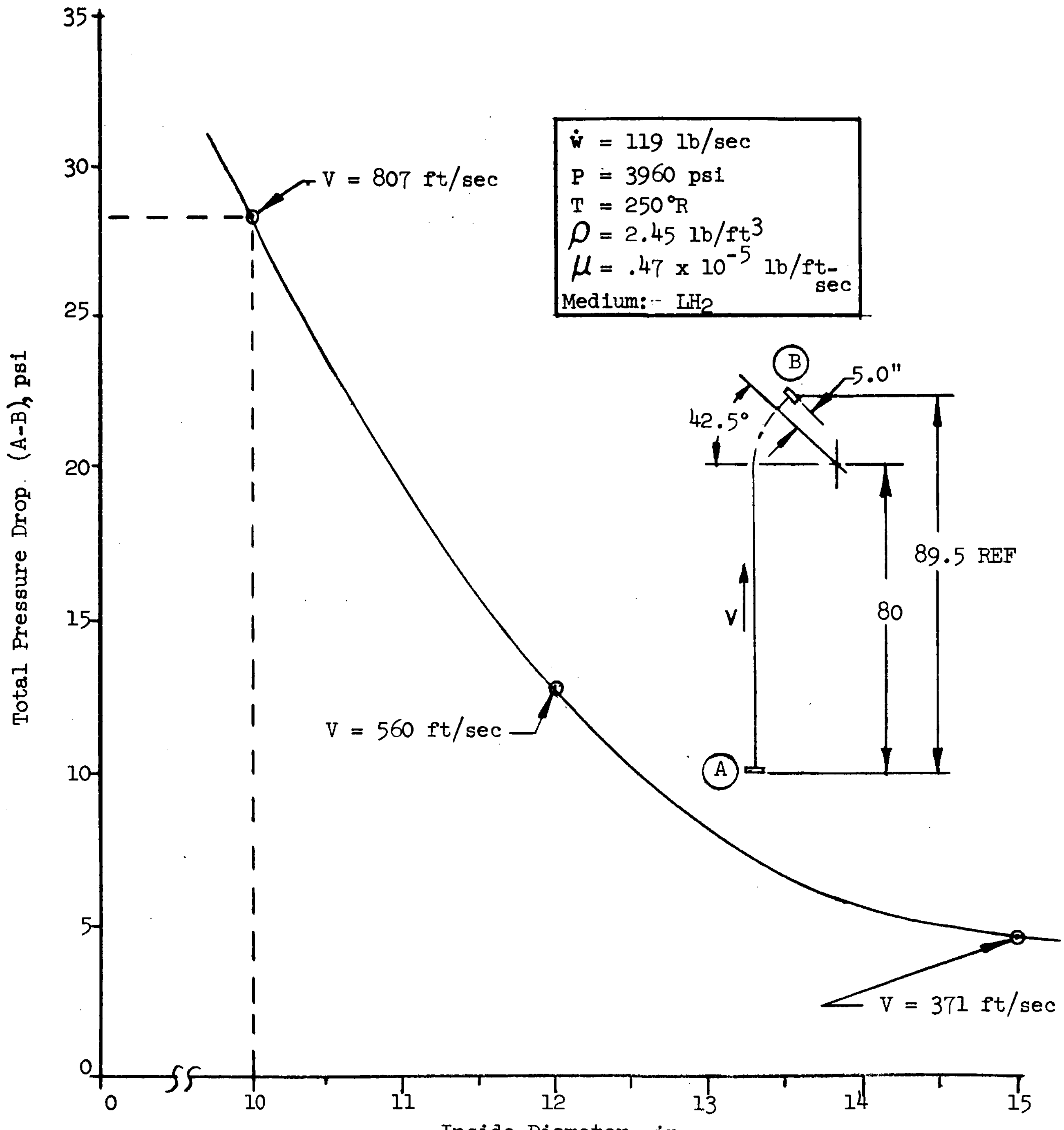
Pressure Drop vs Inside Diameter, Turbine Hot Gas Line, AJ-5 Engine

Figure VIII-B-16



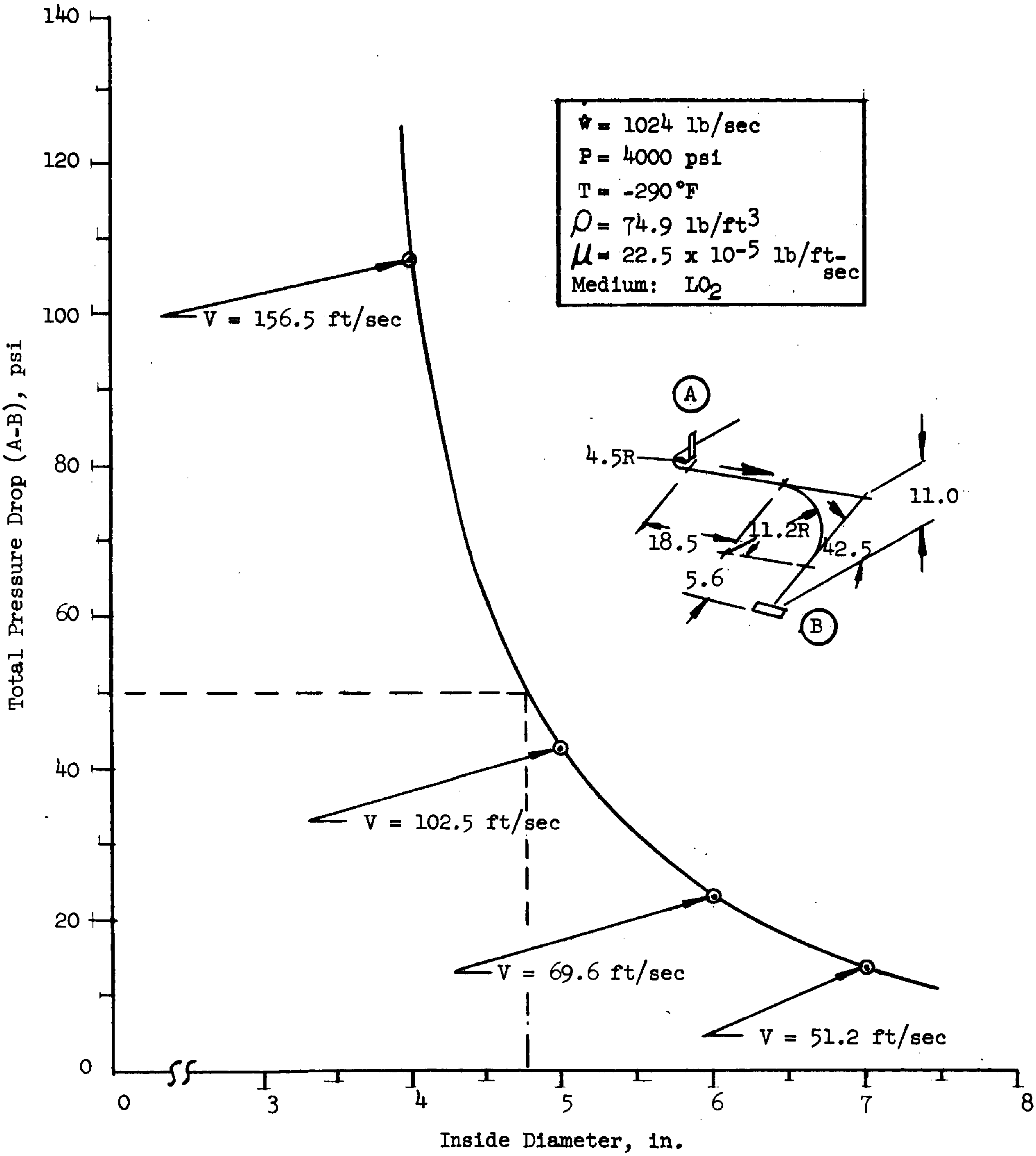
Pressure Drop vs Inside Diameter, Turbine Hot Gas Line - Flow Divided Between Two Lines, AJ-5 Engine

Figure VIII-B-17



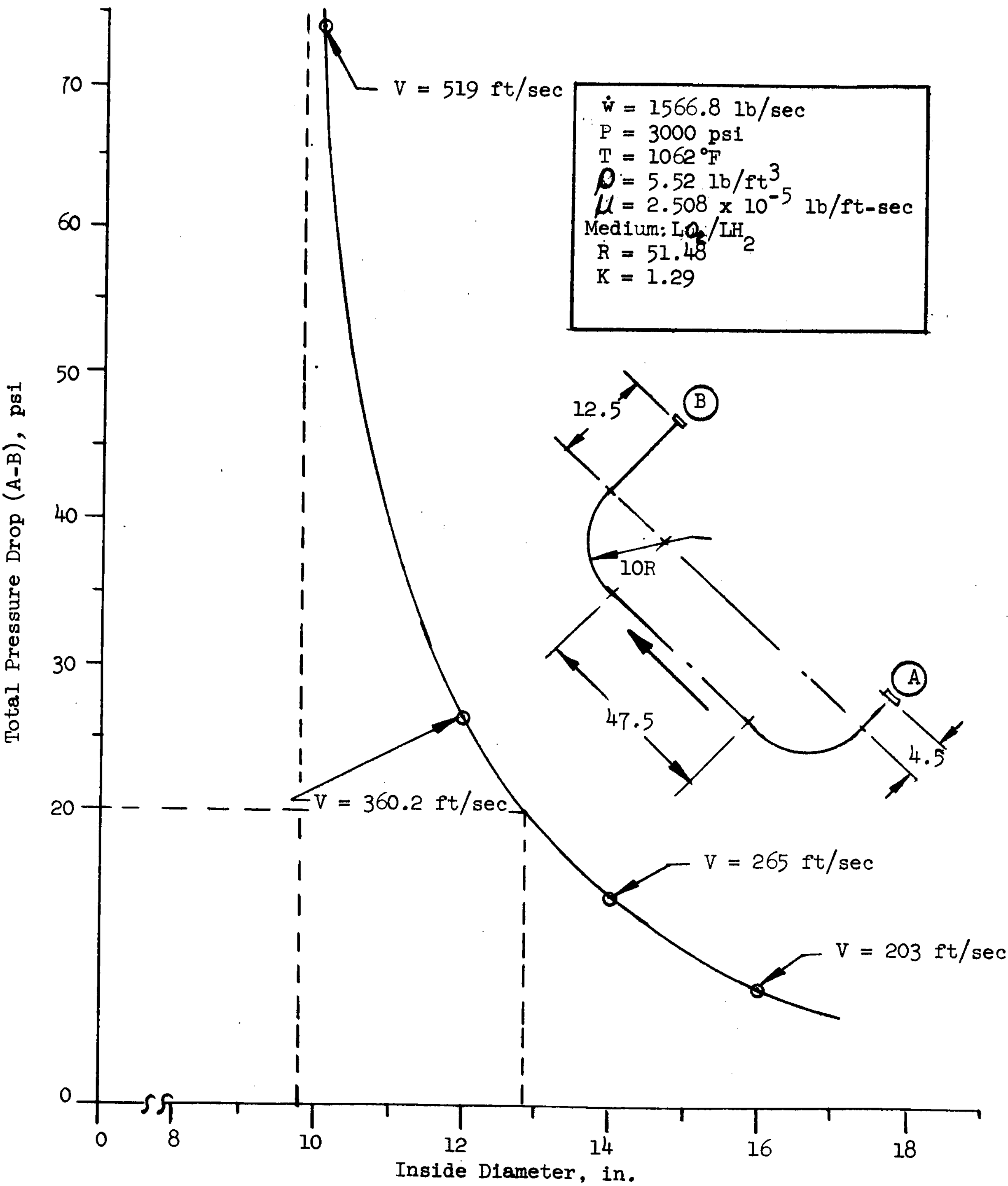
Pressure Drop vs Inside Diameter, Fuel Line to Primary Combustor AJ-6 Engine

Figure VIII-B-18



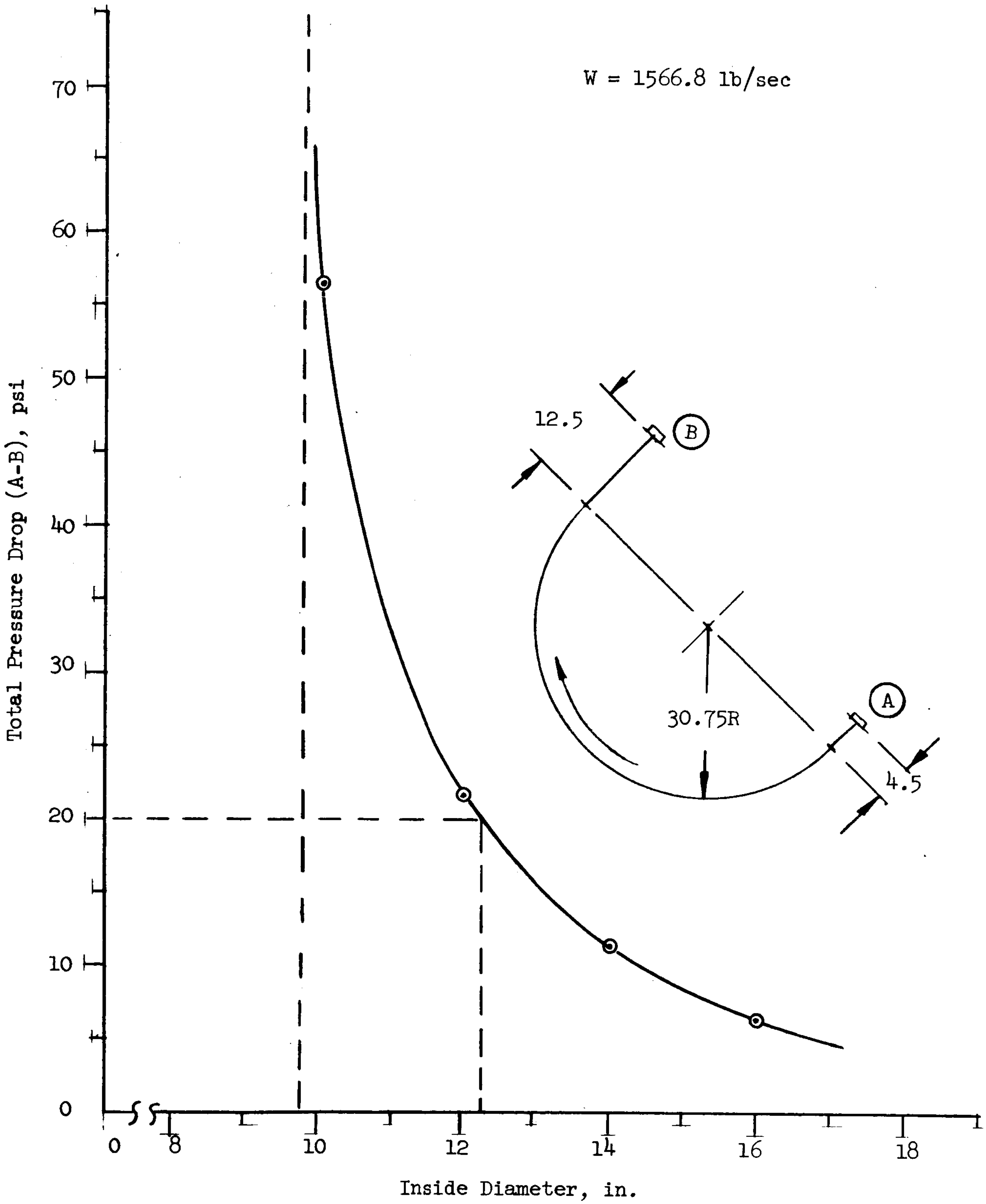
Pressure Drop vs Inside Diameter, Oxidizer Line to Primary Combustor, AJ-6 Engine

Figure VIII-B-19



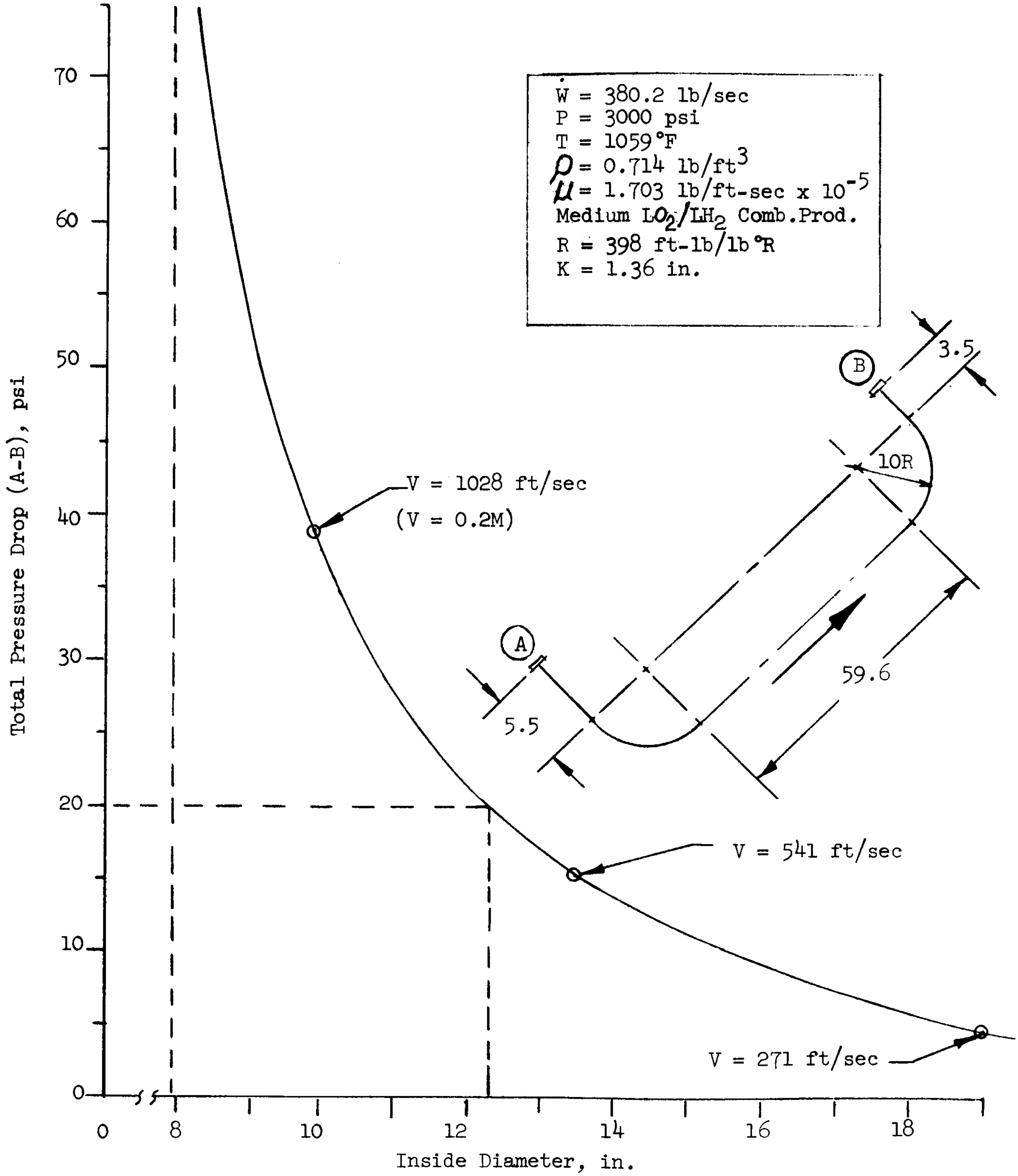
Pressure Drop vs Inside Diameter, O_2-Rich Hot Gas Line to Injector AJ-8 Engine (Two Elbows)

Figure VIII-B-20



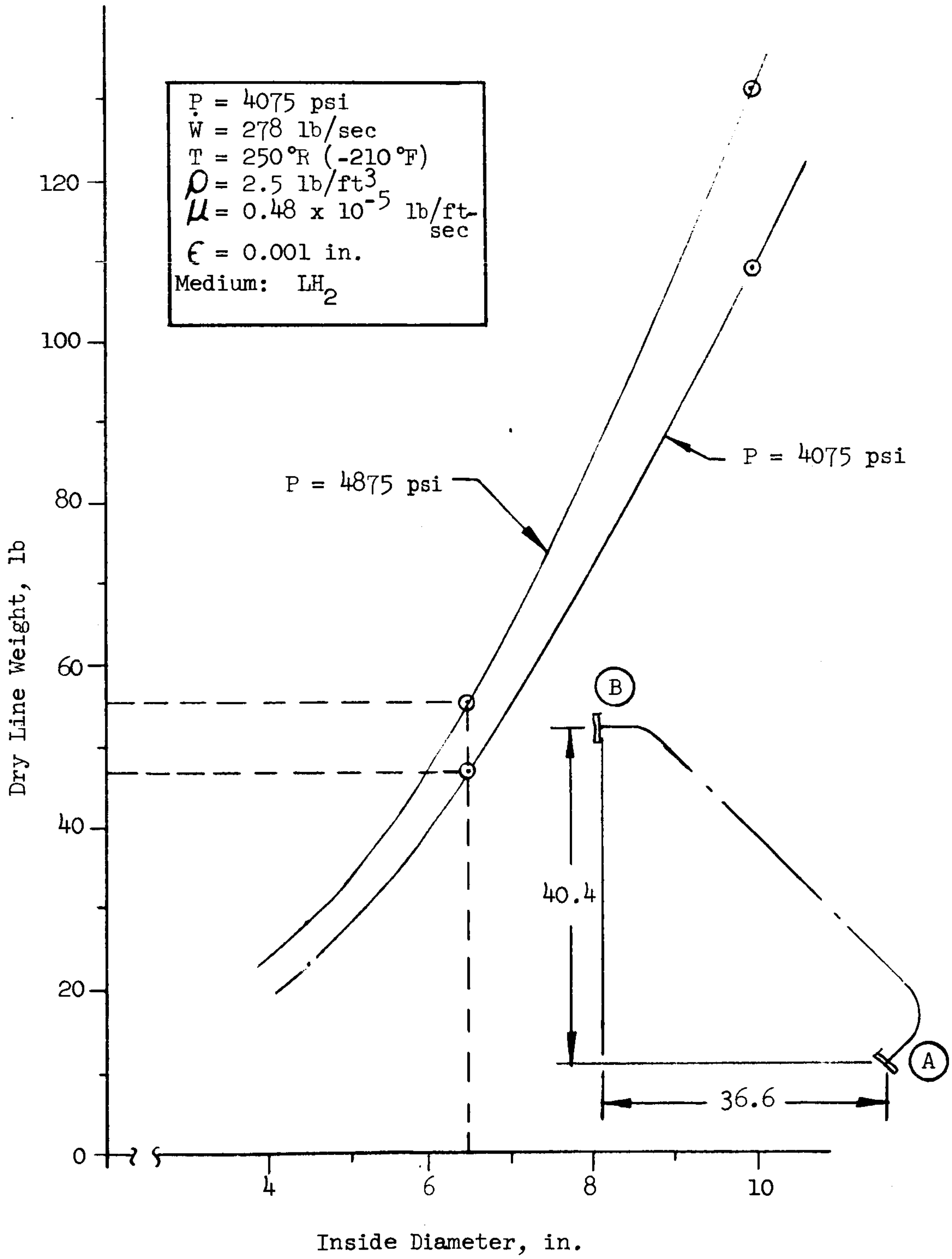
Pressure Drop vs Inside Diameter, O_2-Rich Hot Gas Line to Injector AJ-8 Engine (One Large Radius Bend)

Figure VIII-B-21



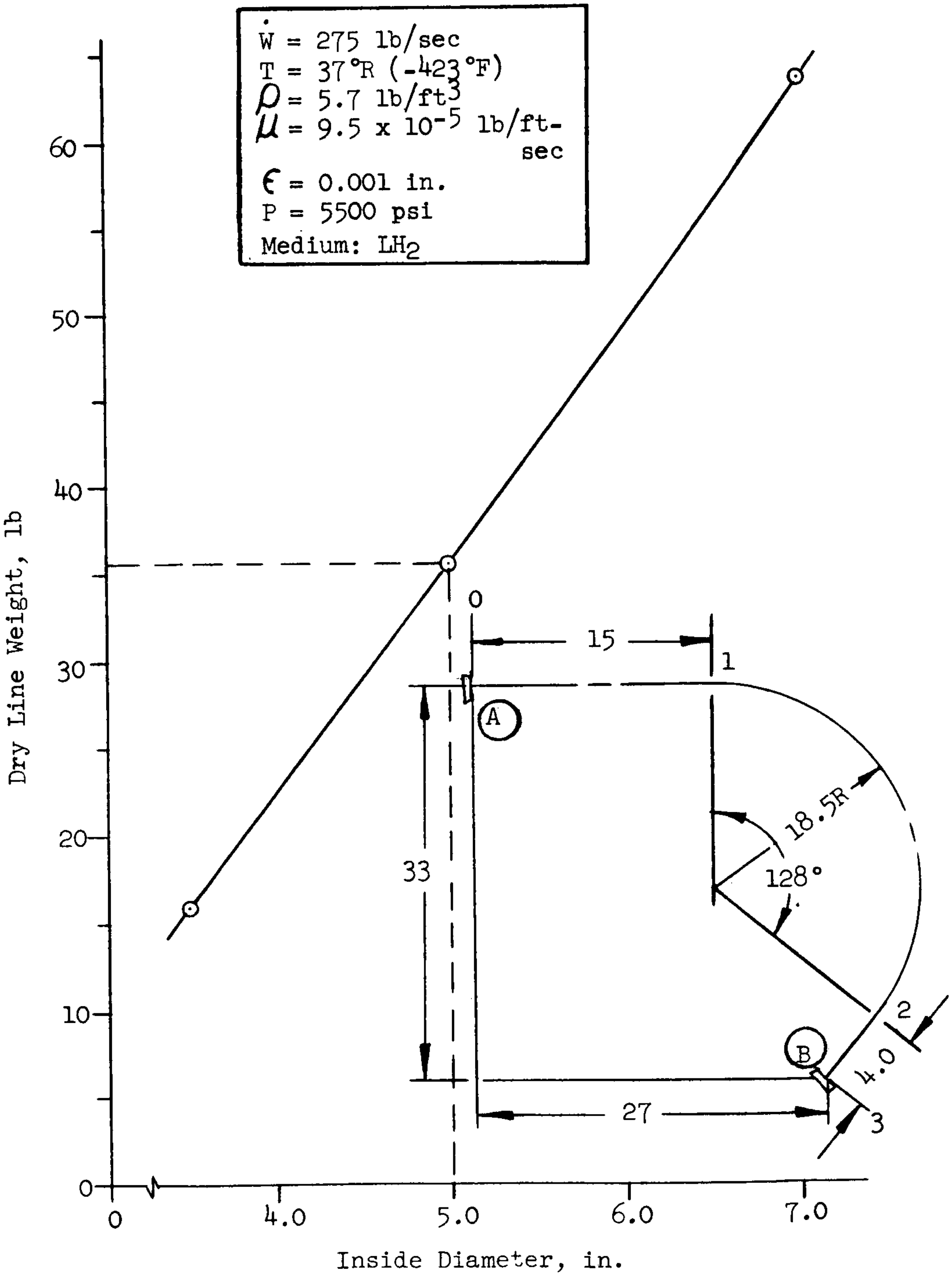
Pressure Drop vs Inside Diameter, H_2-Rich Hot Gas Line to Injector AJ-8 Engine

Figure VIII-B-22



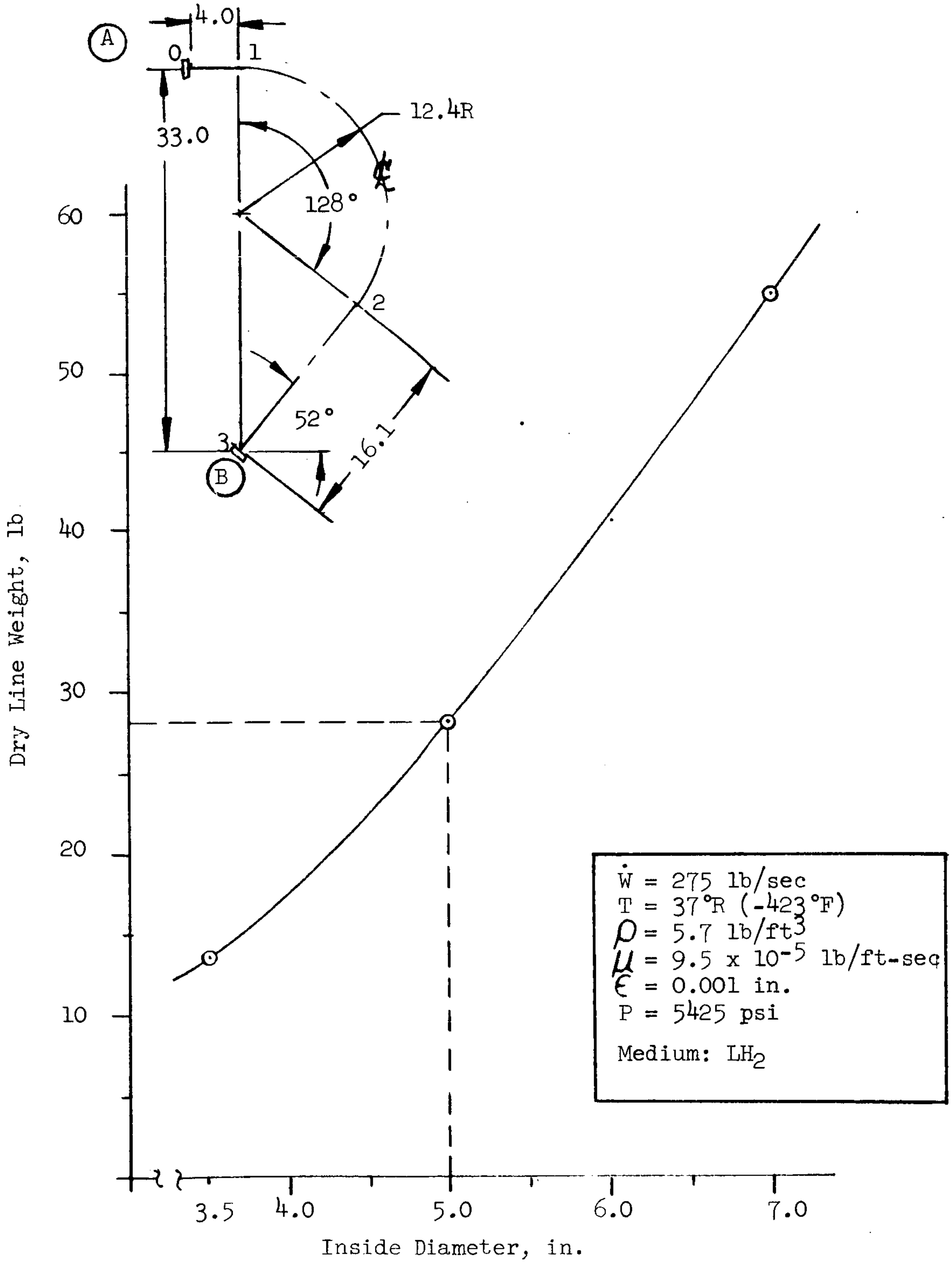
Dry Line Weight vs Inside Diameter, Fuel Line to Primary Combustor for the AJ-1 Engine

Figure VIII-B-23



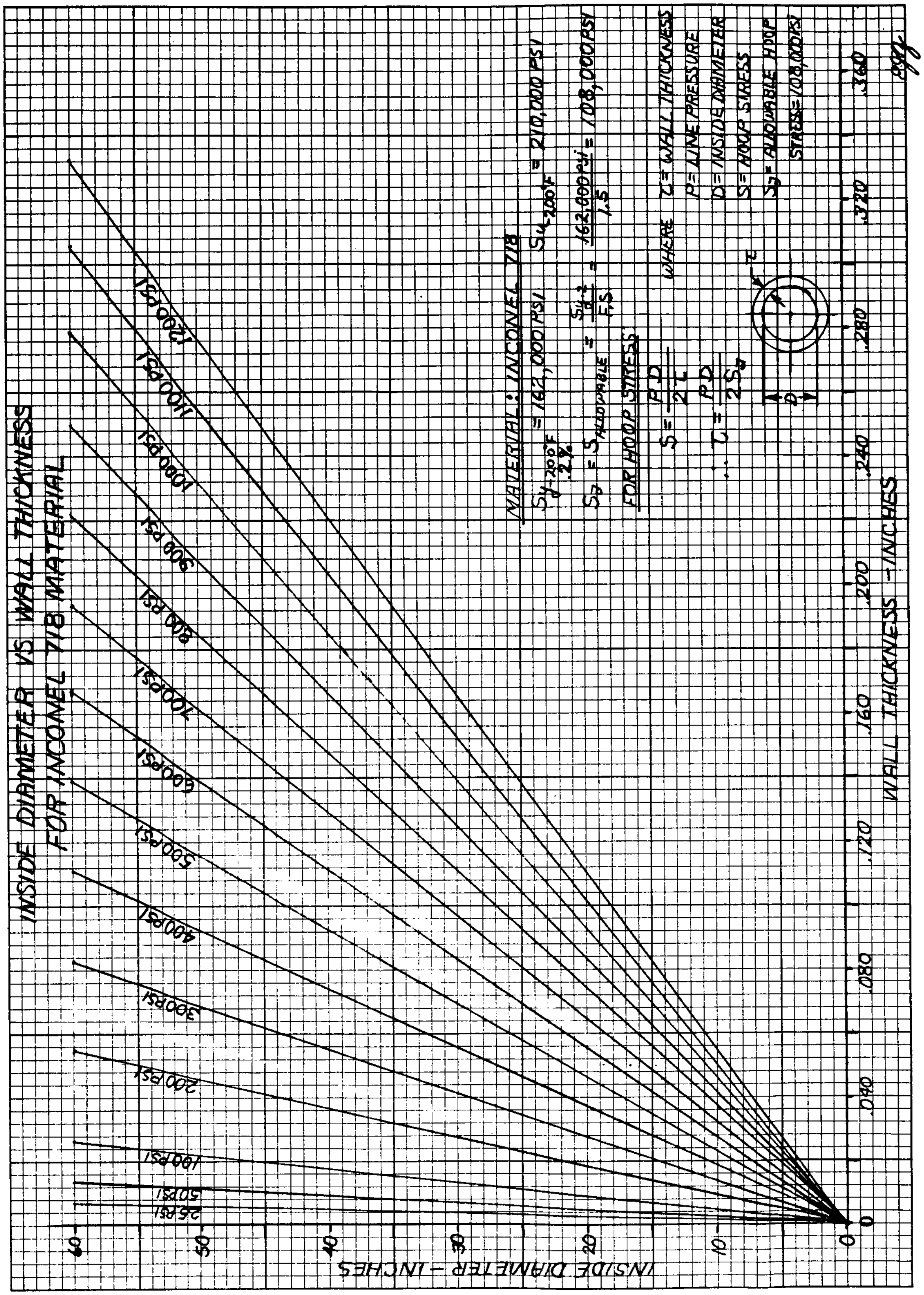
Dry Line Weight vs Inside Diameter, Fuel Line to Secondary Combustor for the AJ-1 Engine (without Valves)

Figure VIII-B-24



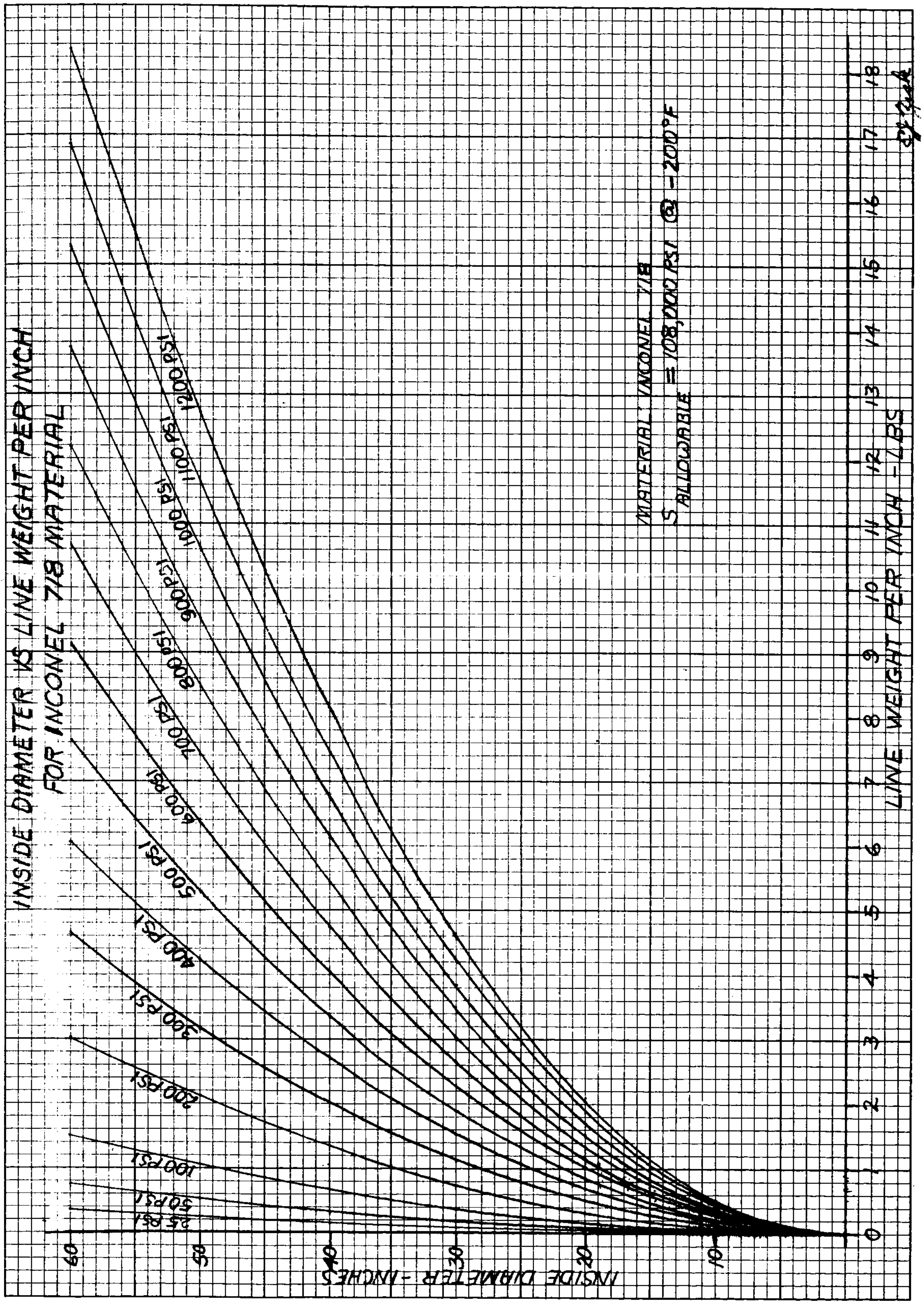
Dry Line Weight vs Inside Diameter, Fuel Line to Secondary Combustor for the AJ-1 Engine (with Valve)

Figure VIII-B-25



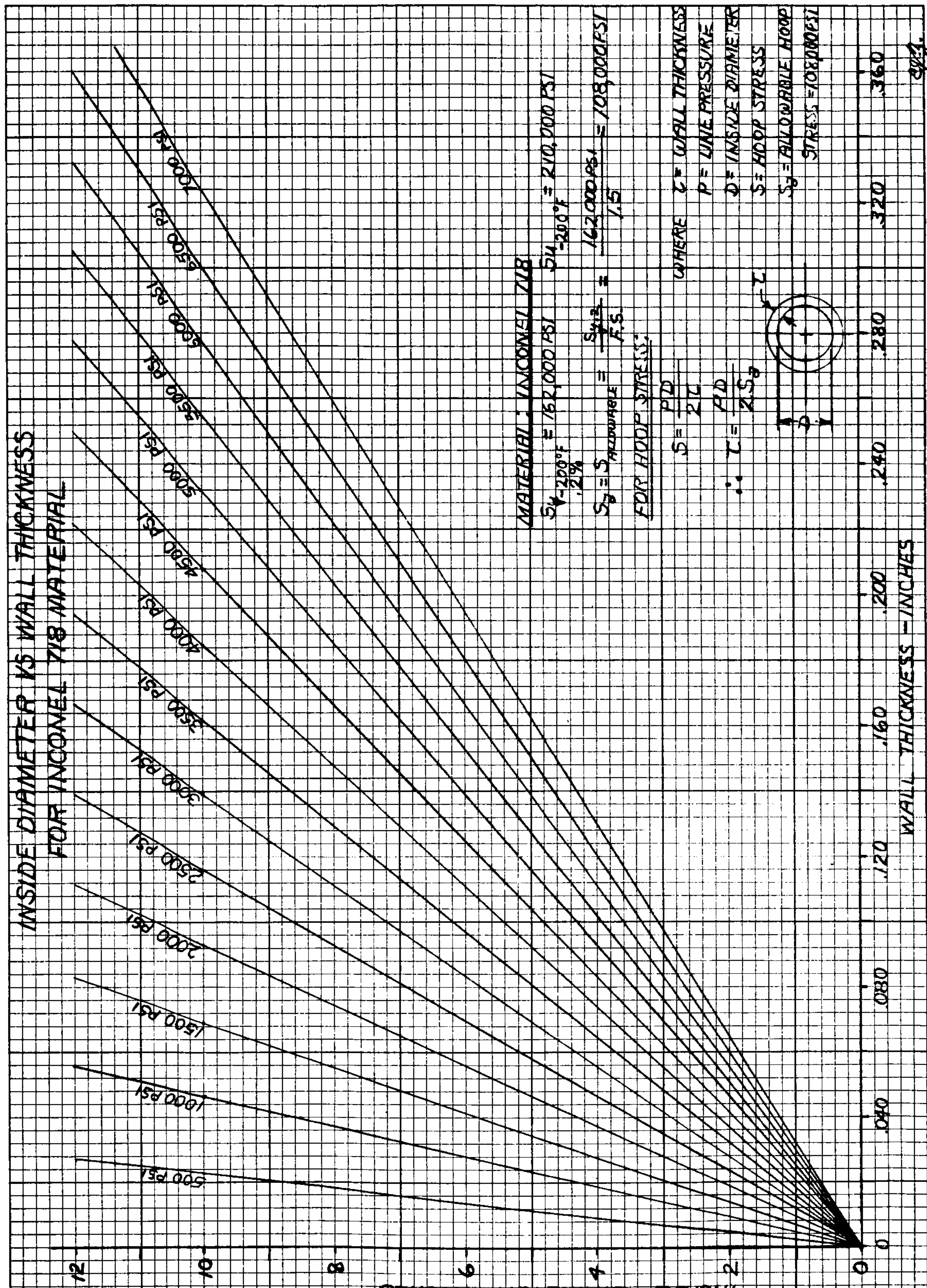
Inside Diamter vs Wall Thickness for Inconel 718 Material, 25 to 1200 psi

Figure VIII-B-26



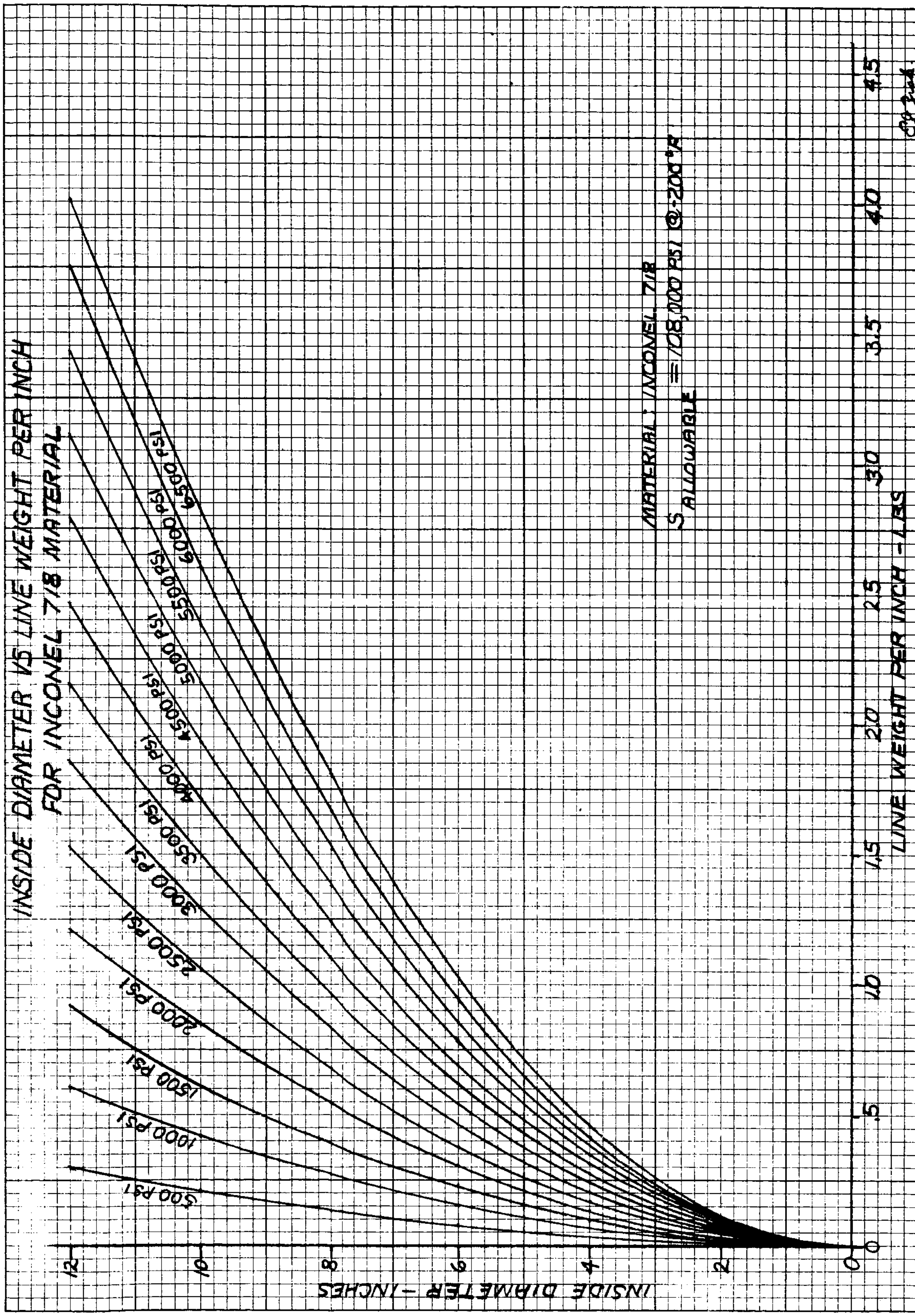
Inside Diameter vs Line Weight for Inconel 718 Material, 25 to 1200 psi

Figure VIII-B-27



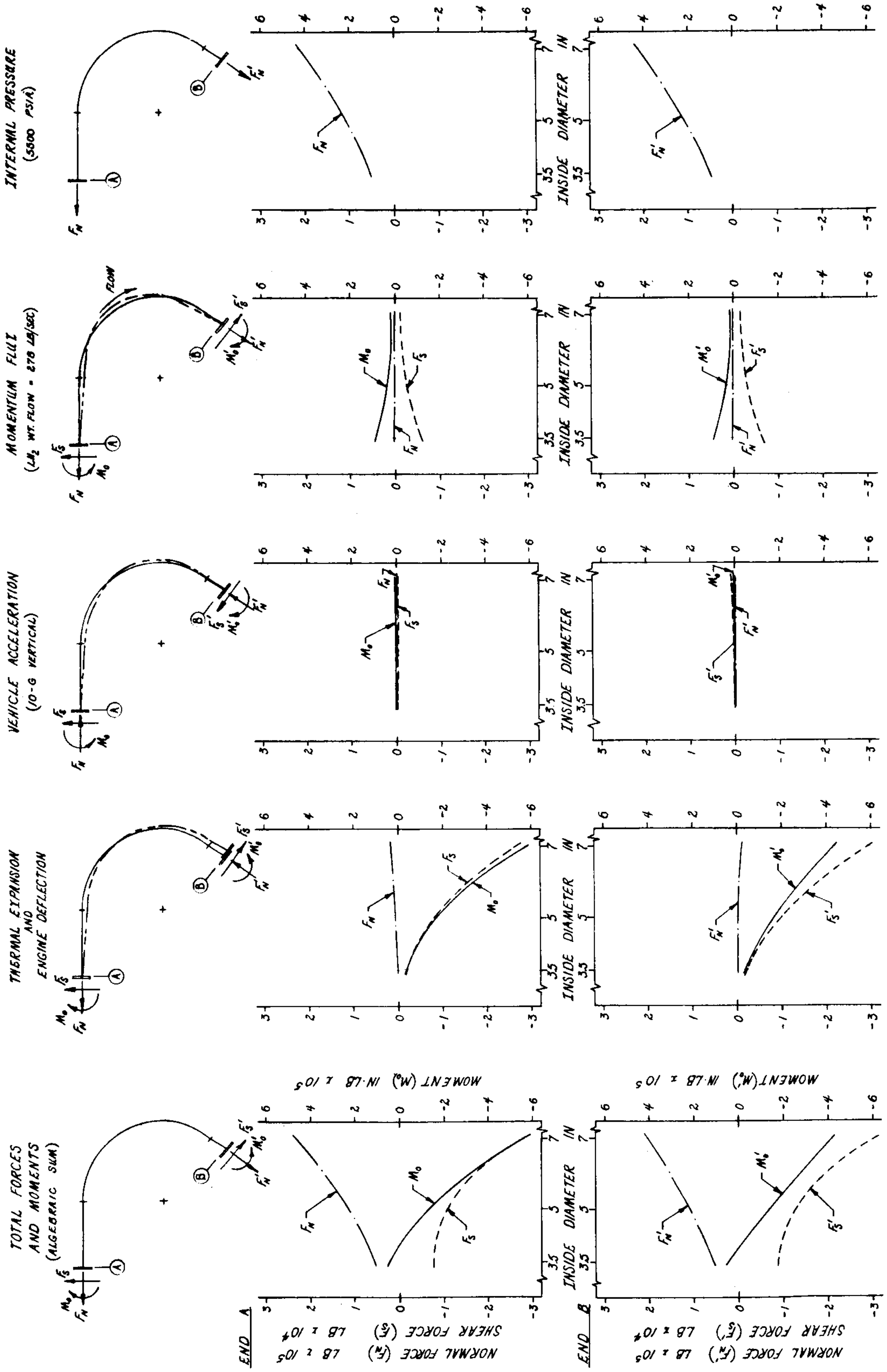
Inside Diameter vs Wall Thickness for Inconel 718 Material, 500 to 7000 psi

Figure VIII-B-28



Inside Diameter vs Line Weight for Inconel 718 Material, 500 to 6500 psi

Figure VIII-B-29



Structural Analysis of AJ-1 Fuel Pump Discharge Line to Secondary Combustor

Figure VIII-B-30

VIII, Lines, Ducts, and Flexible Joints (cont.)

C. FLEXIBLE BELLOWS JOINTS

This section discusses the functional requirements of bellows joints for large, high-pressure rocket engines, the basic design considerations associated with bellows and restraint devices, and the state-of-the-art relative to design and fabrication of such joints.

1. Functional Considerations

Flexible joints are used for the following purposes:

a. Reduce line end-reaction forces (Sections VIII, A and B)

b. Reduce stress in the line

c. Allow for tolerance stackup at assembly (or avoid necessity for precision assembly)

d. Permit installation of parts in tight quarters

The sizes and pressures of lines, as discussed previously, apply equally to flexible joints. Review of the engine design studies indicates that engines of higher thrust level and pressure tend to require lines of increasingly smaller length-to-diameter ratios. This trend not only reduces the inherent flexibility of the line, but provides proportionately less length to accomodate flexible joints and demands larger gimbaling capabilities of these joints. To further complicate the bellows design problem, the high pressure and relatively high velocity of the fluid may require such increases in the cross-sectional area of internal-restraint devices that the bellows inside diameter must be increased to maintain reasonable fluid pressure drop. This increase in diameter will significantly increase the force that the restraint must withstand.

VIII, C, Flexible Bellows Joints (cont.)

The anticipated bellows size and pressure requirements for these large, high-pressure engines are indicated graphically in Figure VIII-C-1. The summary of lines, Table VIII-A-1, gives representative nominal-line diameters as well as fluid pressures and temperatures for a variety of engine lines, any of which might well be expected to require bellows joints.

These figures represent only the gross parameters of size and pressure. The actual definition of the severity of a bellows design requirement must also include gimbaling spring rate, cycle life, temperature, and allowable pressure drop.

A sample calculation of the gimbaling or bending required by the AJ-1 fuel-discharge line on which the detailed structural analysis was performed (Section VIII, B, 3) showed that not more than 6° would be needed. To estimate a more severe requirement for a large engine, a very conservative analysis was performed on the turbine hot-gas line of the AJ-5 engine (Figure VIII-B-16), which resulted in a maximum angulation requirement of 12°. Designing the line with optimum bellows location would reduce this requirement to 7°. These values are essentially the same as present bellows gimbaling requirements.

Spring-rate requirements will almost certainly be based on the maximum reaction load the mating component can tolerate rather than stress limitations within the line itself. While high-pressure pumps, valves, and combustion chambers will of necessity be stronger than low-pressure units, there will still remain severe restrictions on the magnitude of any line end reaction remain. The primary reason for these restrictions is to avoid structural distortions that could result in rubbing of pump impellers or turbine blades, or binding or leaking in valves. While the high operating pressures will dictate the use of higher strength materials for housings, etc., the resulting structures will not necessarily be proportionally more resistant to distortion. This is because distortion, or bending, is a function of the EI product rather than the stress level.

Cycle life, temperature, and allowable pressure-drop requirements are not expected to be any more severe than for present applications.

2. Basic Bellows Joint Design Considerations

a. Bellows

Flexible joints for high-pressure systems usually consist of a bellows with an axial restraint device. Bellows are divided into two types, welded bellows and formed bellows, Figure VIII-C-2. Welded bellows are used where motion is primarily axial with limited lateral offset and angular deflection. The spring rate of welded bellows is generally low and the bellows flexibility is high. However, their application has been primarily in the fields of positive expulsion and flexible seals. Of the two basic types of bellows, formed and welded, only formed bellows normally lend themselves to transfer-line flexible-joint application. While significant advances have been made recently in the analytical design techniques for welded bellows, especially relative to positive-expulsion device application,* the information is of relatively little value in the analysis of high-pressure formed bellows. Welded bellows are fabricated in single or double ply and consist of a series of formed discs welded at the inside and outside diameter of each convolution (Figure VIII-C-2). Single-ply welded bellows have been fabricated for inside diameters up to 3.5 in. for pressures up to 10,000 psi.

The formed bellows is best suited for liquid and hot-gas line application in rocket engines because of its ability to meet angular deflection, diameter, and pressure requirements. Spring rates of formed bellows are generally higher than welded bellows; however, the added stiffness contributes to the structural stability of the formed bellows at large angular deflections.

*Study of Zero-Gravity Positive Expulsion Techniques, Bell Aerosystems Report 8230-933004, June 1963.

VIII, C, Flexible Bellows Joints (cont.)

Formed bellows are usually hydraulically or roll-formed. Hydraulic forming is preferred because it produces more uniform corrugations with a minimum thinning of the bellows material. Convolutions are formed simultaneously during hydraulic forming with 10%, or less, wall thinning. The entire process is accomplished at room temperature. With roll forming, convolutions are formed one at a time with 10%, or less, wall thinning becoming more difficult to attain.

The formed bellows can be made of single- and multiple-ply construction. The multiple-ply or laminated construction has been used to obtain maximum flexibility at high pressures. Figures VIII-C-3 through -6 show a 3.5-in.-dia, 8400-psi proof-pressure bellows made of 10 plies of 0.008-in. Type 347 stainless steel, with high tensile-strength root rings and wire-mesh external reinforcement.

Root rings or equalizing rings as they are sometimes called are used to increase the amount of hoop stress a joint can sustain. Therefore, thinner bellows walls and a lower spring rate can be used. The external rings also prevent lateral buckling of the bellows by providing support at the inner edge.* The root-support rings are positioned on the outside surface of each inside convolution of a bellows.

b. Restraint Devices

The solution of all problems associated with actual bellows design will not result in efficient designs for high-pressure flexible joints if the improvement of the necessary axial-restraint device is not considered. With the exception of bellows joints designed expressly for axial motion or for single bellows in extremely short lines, practically all rocket-engine flexible joints require a pivoting axial-restraint device.

*Zallea Expansion Joints, Catalog No. 56, 1956 Zallea Bros., Wilmington, Delaware.

VIII, C, Flexible Bellows Joints (cont.)

The axial-restraint device is an excellent example of a design problem for which a solution can almost always be found, although possibly at a severe sacrifice in weight and envelope.

The design of axial-restraint devices for high-pressure flexible joints increases in difficulty at a rate directly proportional to the increase in pressure. Design criteria for present internal-restraint-device concepts indicate that the cross sectional area of the device must increase proportionately with the increase in pressure. Preliminary design studies have shown the blockage to free stream-flow for a 10-in.-dia joint to increase from 18 to 48% with an increase in proof pressure from 2500 to 6500 psi. For the same weight flow, the pressure drop was more than quadrupled.

The increased fluid velocity around the restraint mechanism results not only in increased pressure drop through the joint, but also in additional load requirements due to fluid drag on the restraint components, and increased likelihood that bellows liners will be required.

The increase in blockage area with most designs becomes far more severe when the joint is designed for high-temperature service, because of the reduction in load-carrying capacity of most bearing materials at elevated temperature. This indicates that appreciable improvement may be realized with the evaluation and selection of optimum materials for the internal-restraint bearing surfaces.

The use of external-restraint devices to avoid these problems requires substantial increase in envelope volume as well as a significant increase in overall joint weight.

VIII, C, Flexible Bellows Joints (cont.)

3. Evaluation of the State-of-the-Art

For the initial effort to evaluate the state-of-the-art relative to the manufacture of bellows joints, two inquiries were made. The first consisted of requests for bid from 12 leading bellows manufacturers (Table VIII-C-1) for design and manufacture of three bellows units. The operating requirements for 6500-psi bellows for use in lines of 3.62-in. (equivalent to an existing Titan line size), 5-in. and 10-in. internal diameter were incorporated on individual drawings that accompanied the bids. The 12 bellows manufacturers were invited to suggest new fabrication techniques, new materials, and new bellows concepts that might solve the problems associated with the stringent operating requirements. Nine of the 12 vendors responded: six submitted no bids; one submitted a bid for a 3.62-in. ID welded bellows only; one submitted a bid for 3.62-in. ID and a 5.00-in. ID formed bellows; and one expressed optimism that bellows up through 10-in. ID could be developed.

An evaluation of these replies showed this approach to be a relatively poor means of establishing manufacturers' capabilities. Many factors, such as facility limitations, shop or engineering work load, delivery time, etc., were presented as reasons for no bid replies and may also have accounted for the lack of response from three of those queried. One manufacturer, who declined to bid, later indicated that this was due to lack of facilities to form heavy-walled bellows at such small diameters and had the request included a bellows between 10 in. and 20 in. in diameter they would probably have bid.

The second inquiry consisted of a letter to 17 bellows manufacturers requesting an indication of the proof pressures for which bellows of specified diameters have been manufactured. Only six companies provided the desired information, which is summarized in Table VIII-C-2 and plotted in Figure VIII-B-30.

VIII, C, Flexible Bellows Joints (cont.)

It should be noted that a 3.5-in.-dia, 8400-psi proof-pressure bellows represents the highest pressure shown. This unit has been developed for an advanced liquid-rocket-engine program at Aerojet-General and is not a flight-weight design. Figures VIII-C-3 through -6 show the completed bellows joint and sections through an early development model of the same bellows, which has a distorted convolution shape resulting from instability experienced near the proof pressure. This joint uses both external root rings and a wire mesh over the outside diameter of the convolutions to retain stability. The final design utilizes teardrop-shaped root rings, because one of the circular cross-section rings shown on the early model failed. Restraint is external and is designed for shear, or lateral offset. This unit has demonstrated a spring rate requiring approximately 7000 lb to deflect one flange 0.5 in. relative to the other.

The present state-of-the-art is also indicated by the results of effort in this program to determine criteria for predicting the effect of high pressure on the angular spring rate of a hinged bellows joint. Two bellows manufacturers have indicated that bellows spring rate (measured unpressurized) may be considered to be roughly directly proportional to design proof pressure, with the convoluted length and mean diameter held constant. This appears reasonable, although an increase in proof pressure usually results in a total design revision with changes in number of plies, ply thickness, and convolution configuration. (It may be readily appreciated that a change in wall thickness will alter the convolution proportions even through basic dimensions such as pitch and internal diameter remain constant.)

Daniels* and Matheny** each present expressions for relating axial spring rate to various functions of the cube of the wall thickness, with no consideration of the number of plies involved, but taking convolution shape into consideration.

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*C. M. Daniels, "Designing for Duct Flexibility with Bellows Joints," Machine Design, 147-155, October 15, 1959.

**J. D. Matheny, "Bellows Spring Rate for Seven Typical Convolution Shapes," Machine Design, 137-139, January 4, 1962.

VIII, C, Flexible Bellows Joints (cont.)

Daniels, referencing the work of Freely and Goryl,* also provides an expression for the moment to rotate a hinge joint, again as a function of the cube of the wall thickness. In this instance, no effect of convolution shape is presented, although basic disc dimensions are considered. Daniels does state, however, "For example, if a bellows has a two-ply wall with a combined thickness equal to a single-ply bellows wall, the two-ply bellows will provide almost twice the flexibility of the single-ply bellows. Within its elastic limit, the action of a convoluted bellows is closely related to that of a helical spring, and deflection is proportional to the applied load." In the course of a meeting with representatives of Bell Aerosystems in October of 1963, L. M. Thompson and I. W. Depuy indicated that their experience with welded bellows showed that the addition of more plies to increase wall thickness does not proportionally increase spring rate.

The work of Seide** gives an indication of the effect of pressure on the moment to rotate a hinge bellows joint where the bellows is of multi-ply construction. Seide does not discuss the effect of bellows construction per se, but rather relates the bellows joint to uniform cylindrical beams of rigid, circular cross section. A relationship is derived relating moment to rotate the joint under pressure to the pressure/critical-pressure ratio and the moment required for the unpressurized bellows. Empirical data is shown for a bellows 13.67-in. OD, 12.42-in. ID, 2.23 in. long with eight convolutions of 12 plies 0.005 in. thick, hinged at the center with internal restraint. Agreement of test data with theoretical prediction was not good, but the data itself is of significant interest. As pressure inside the bellows was increased from 0 to 600 psi, the moment required to angulate the bellows increased from 9,520 to 42,200 in. lb/radian (Figure VIII-C-7, taken from Journal of Applied Mechanics, pp. 429-437, September 1960). In explaining the poor correlation of test data with theoretical, the author suggests, "We may conjecture that for the large bellows, the main reason for the poor agreement results from an increase in the stiffness of the pressurized bellows over the unpressurized

*F. J. Freely Jr. and W. M. Goryl, "Stress Studies on Piping Expansion Bellows," Journal of Applied Mechanics paper 44-APM-22.

**P. Seide, "The Effect of Pressure on the Bending Characteristics of an Actuator System," Journal of Applied Mechanics, 429-437, September 1960.

VIII, C, Flexible Bellows Joints (cont.)

bellows because of increased friction between the plies." It is of definite interest to note that the spring rate of this particular unit increased more than four fold as pressure was increased from 0 to 69% of the critical buckling pressure (870 psi).

Bellows design personnel of a major bellows manufacturer indicated some skepticism that the spring rate of a multiple-ply bellows would increase four-fold under pressure as shown by this data.

TABLE VIII-C-1

BELLOWS MANUFACTURERS

	COMPANY	QUOTE REQUESTED	INFORMATION REQUESTED	FORMED BELLOWS	WELDED BELLOWS
1.	Solar Aircraft Co. 2200 Pacific Highway San Diego, California	X	X	X	
2.	Arrowhead Products Div. of Federal-Mogul-Bower Bearings, Inc. 4411 Katella Avenue Los Alamitos, California	X	X	X	
3.	Dunbar Kapple, Inc. of California 1626 Cantinella Avenue Inglewood 3, California	X	X	X	
4.	Aeroquip Corporation, Marman Div. 11214 Exposition Boulevard Los Angeles, California	X	X	X	
5.	Fairchild Camera & Instrument 1700 W. Johnson Avenue El Cajon, California	X	X	X	
6.	Straza Industries 790 Greenfield Drive El Cajon, California	X	X	X	
7.	Flexonics 300 E. Devon Avenue Bartlett, Illinois	X	X	X	
8.	Sealol Incorporated Warwick Industrial Pl. Providence 5, Rhode Island	X	X		X
9.	Aero-Flex Corporation 7982 Othello Street San Diego 11, California	X	X	X	
10.	Stainless Steel Products 2980 N. San Fernando Boulevard Burbank, California	X	X	X	

TABLE VIII-C-1 (cont.)

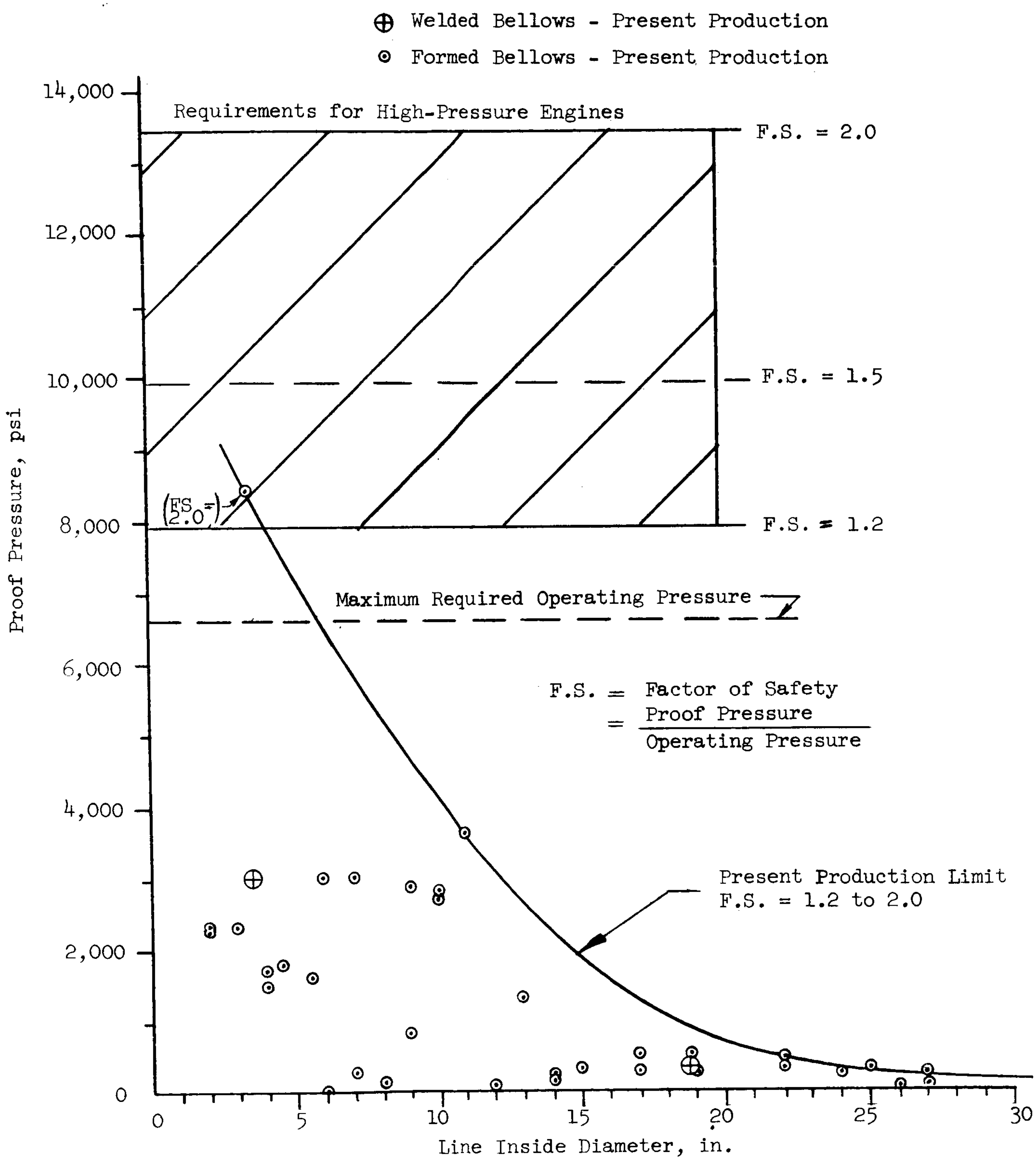
	COMPANY	QUOTE REQUESTED	INFORMATION REQUESTED	FORMED BELLOWS	WELDED BELLOWS
11.	Avica Corporation P. O. Box 180 Newport, Rhode Island	X	X	X	
12.	Flexible Metal Hose Mfg. Co. 777 West 16th Street Costa Mesa, California	X	X	X	
13.	Metal Bellows Corporation 20977 Knapp Street Chatsworth, California	X	X		X
14.	Resistoflex Corporation 301 No. Crescent Street Anaheim, California Western Division of Zallea Brothers Roseland, New Jersey		X	X	
15.	Belfab Corporation P.O. Box 1881 Daytona Beach, Florida		X		X
16.	Hydrodyne Corporation 7350 N. Coldwater Canyon North Hollywood, California			X	
17.	Keflex Manufacturing Division U.S. Flexible Metallic Tubing Co. 63 Main Street San Francisco 5, California			X	
18.	Keller Products Co. 8 Littellroad, New Jersey				X
19.	Anaconda Metal Hose Division Anaconda American Brass Company Waterbury, Conn.			X	
20.	Standard Thompson Corporation 122 Grove Street Waltham 54, Mass.				

TABLE VIII-C-2

PRODUCTION BELLOWS SUMMARY TABLE

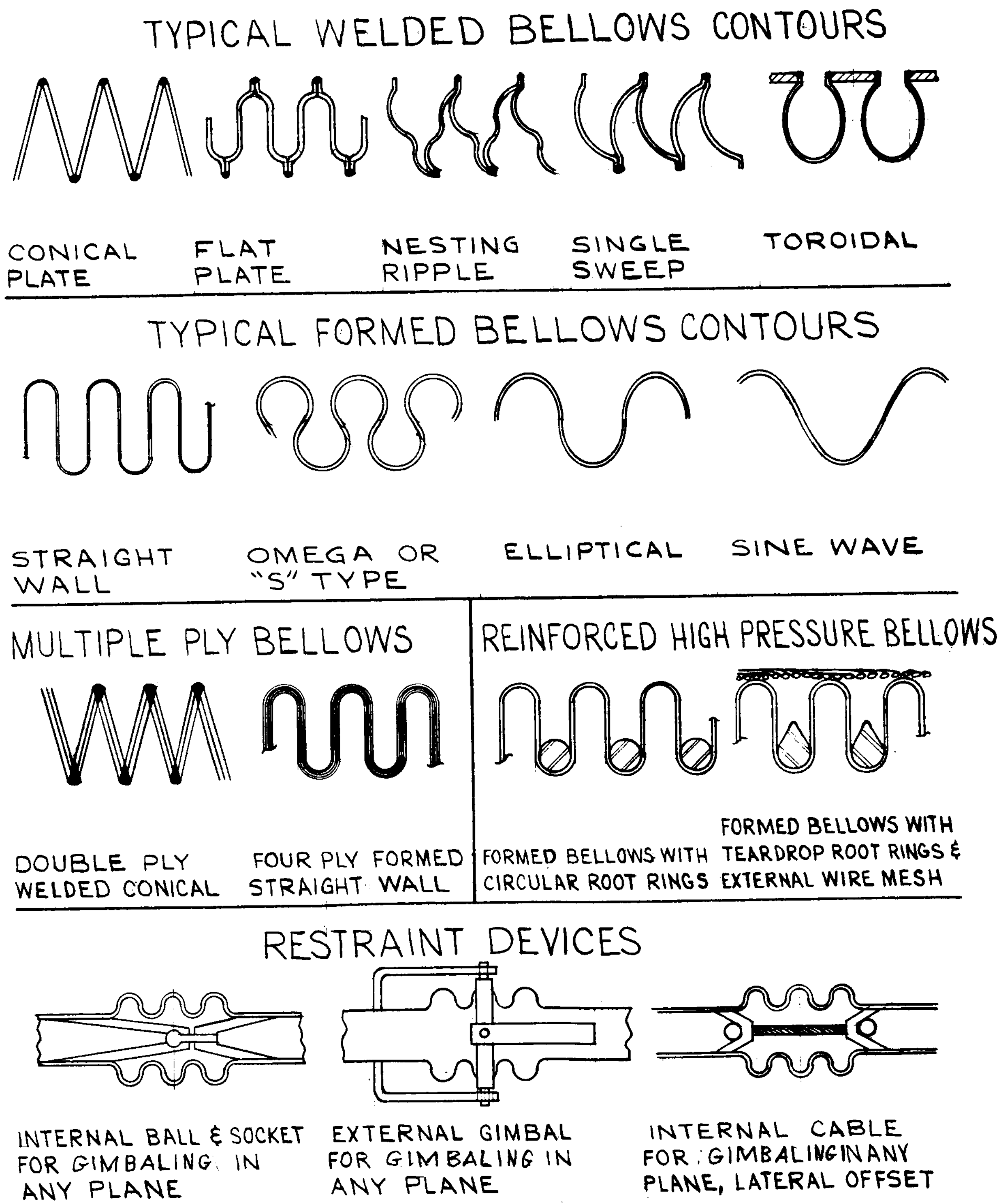
Inside Diameter in.	Proof Pressure psi	Flex Angle, degrees	Bellows Type Formed	Bellows Type Welded	Cycle Life, cycles
2	2300	5	x		5000
2	2250	6	x		
2.062	3000 (double ply)			x	
3	2300	5	x		5000
3.5	8400	*	x		500
4	1700		x		
4	1500	5	x		
4.5	1800	4	x		
5	1600	6	x		
6	20	3	x		4000
6	3000	3	x		
7	250		x		20,000
7	3000	5	x		2000
8	160	2	x		10,000
9	800	6-1/2	x		500
9	2850	5	x		2000
10	2700		x		500
10	2800	8	x		3000
11	3600		x		3000
12	103		x		893
13	1300	7	x		
14	210	15	x		4000
14	150	17	x		1750
15	300		x		
17	500	9-1/4	x		700
17	280	(± 9 in. axial)	x		900
18.80	493	16	x		600
18.75	300			x	
19	225	3	x		500
19	280	16	x		1800
22	405	12	x		1000
22	280	12	x		1100
24	200	(1.75 in. axial)	x		2000
25	280		x		900
26	30		x		2500
27	30		x		2500
27	200	1.62	x		1600
40	40		x		4500

* Designed for shear, or lateral offset of 0.5 in. under 7000-lb load. See Figures VIII-C-3 through -6.



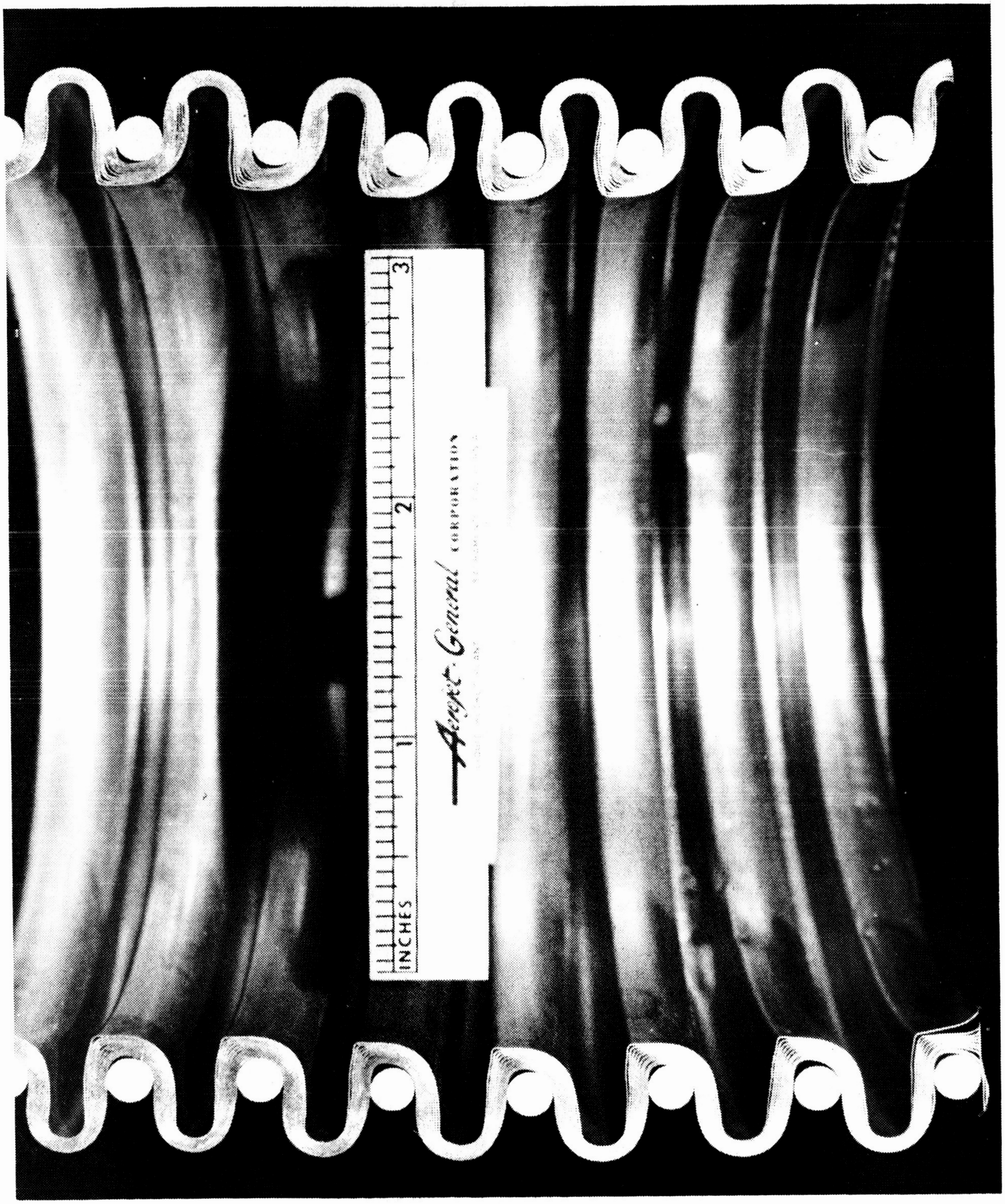
Bellows Size and Pressure

Figure VIII-C-1



Typical Bellows Description

Figure VIII-C-2



3.5-in. 8400 psi Proof Bellows Section

Figure VIII-C-3

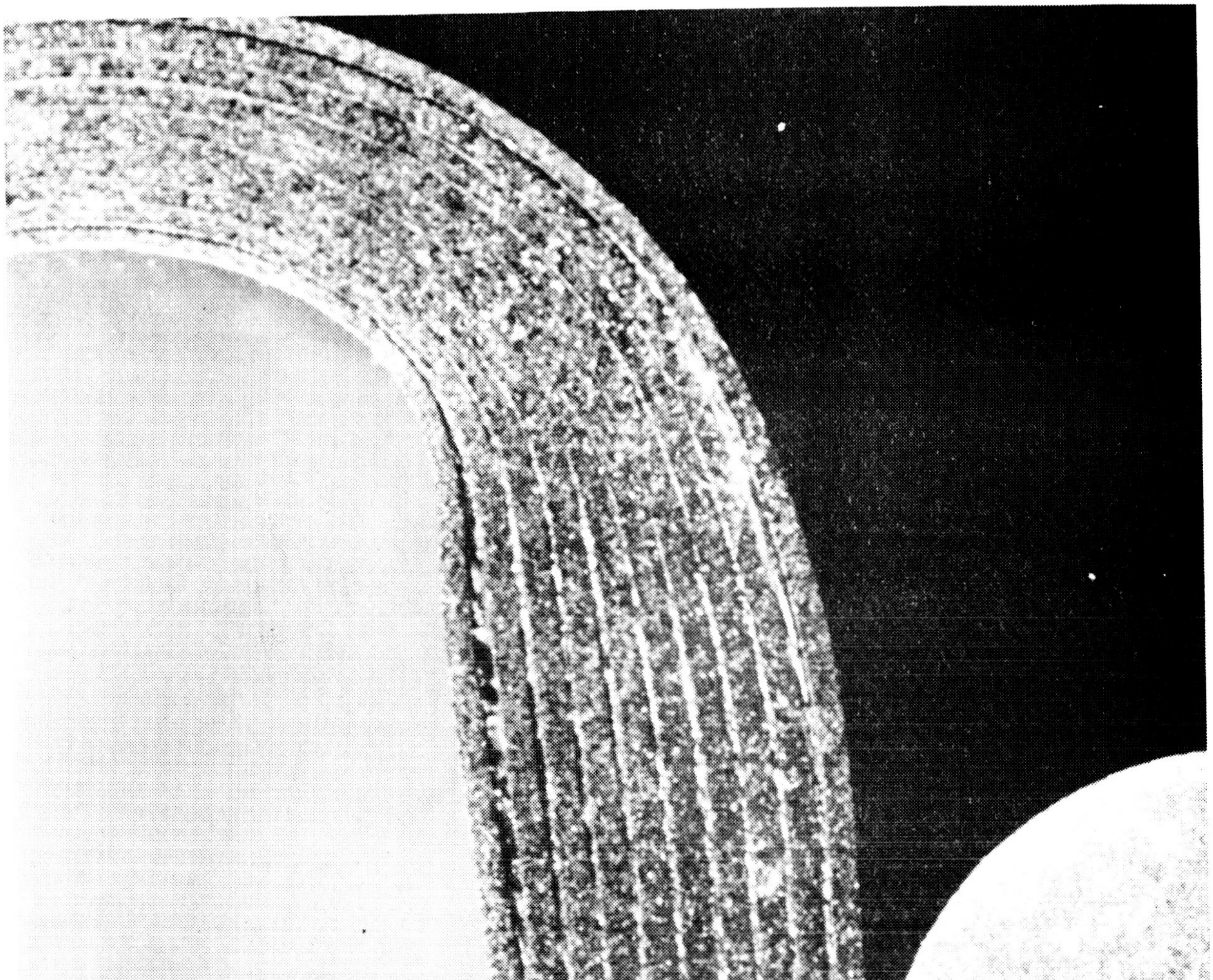
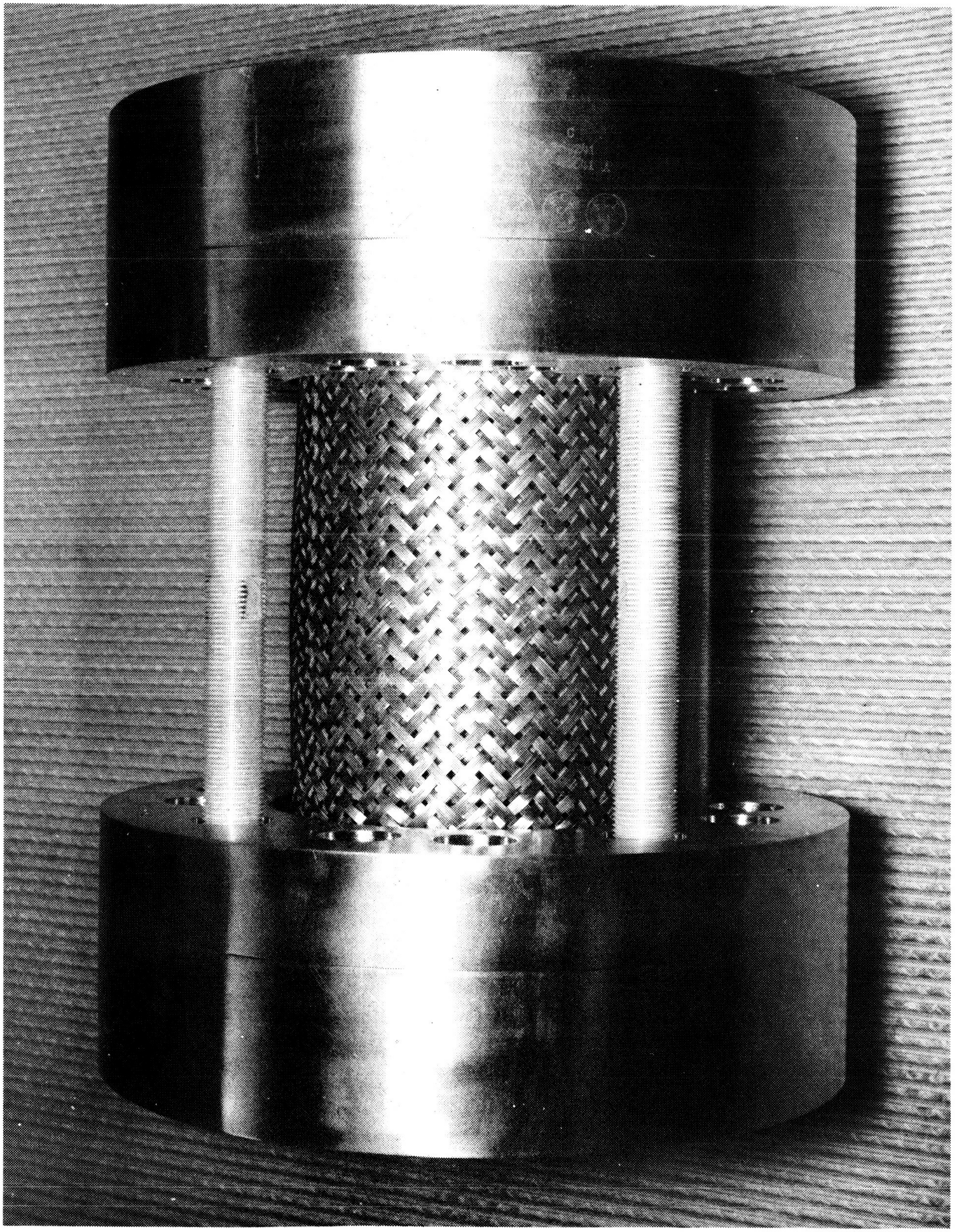


Figure VIII-C-4

Close-up of Bellows Section



3.5-in. 8400 psi Proof Bellows Joint (Exterior)

Figure VIII-C-5

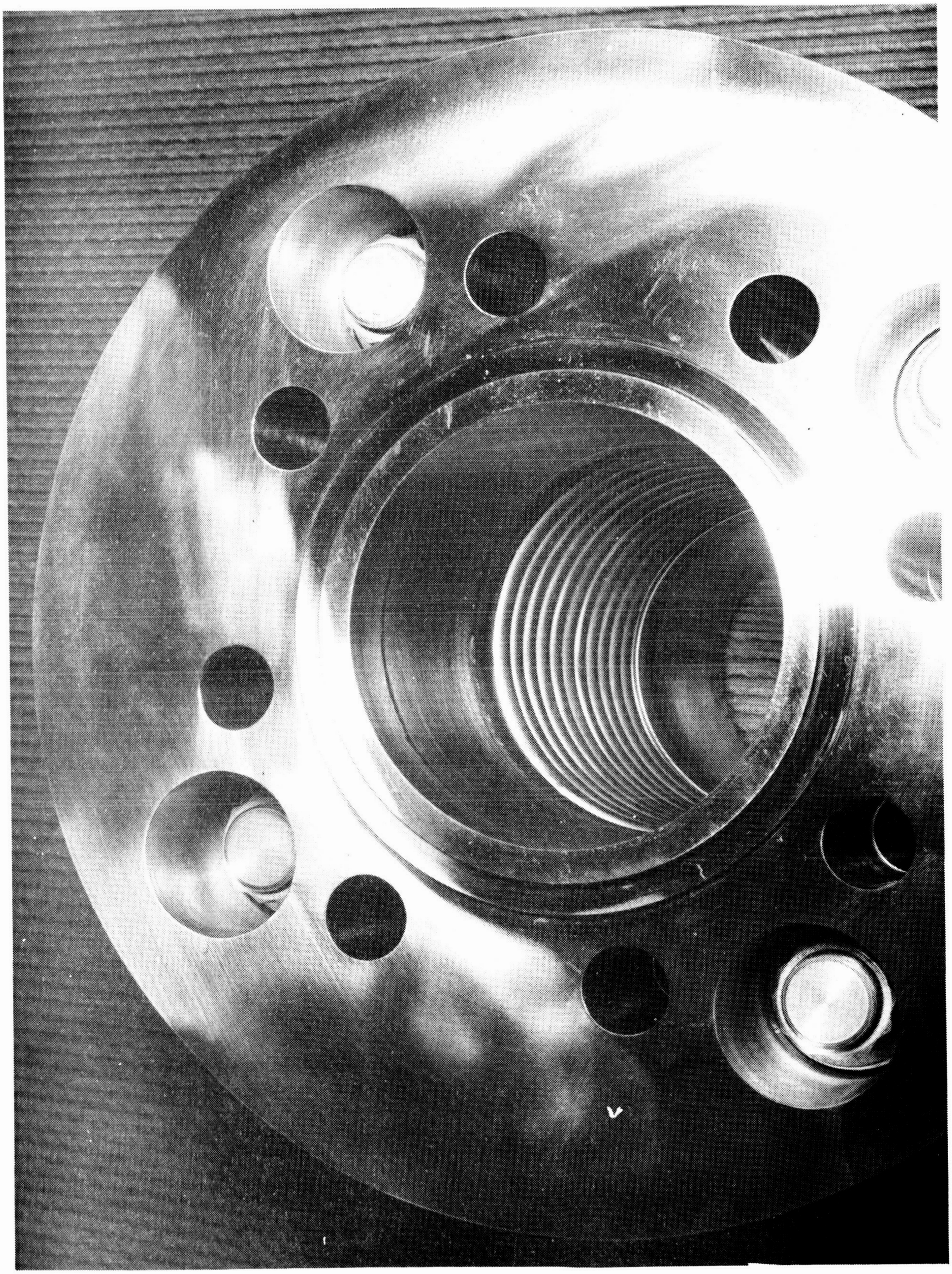
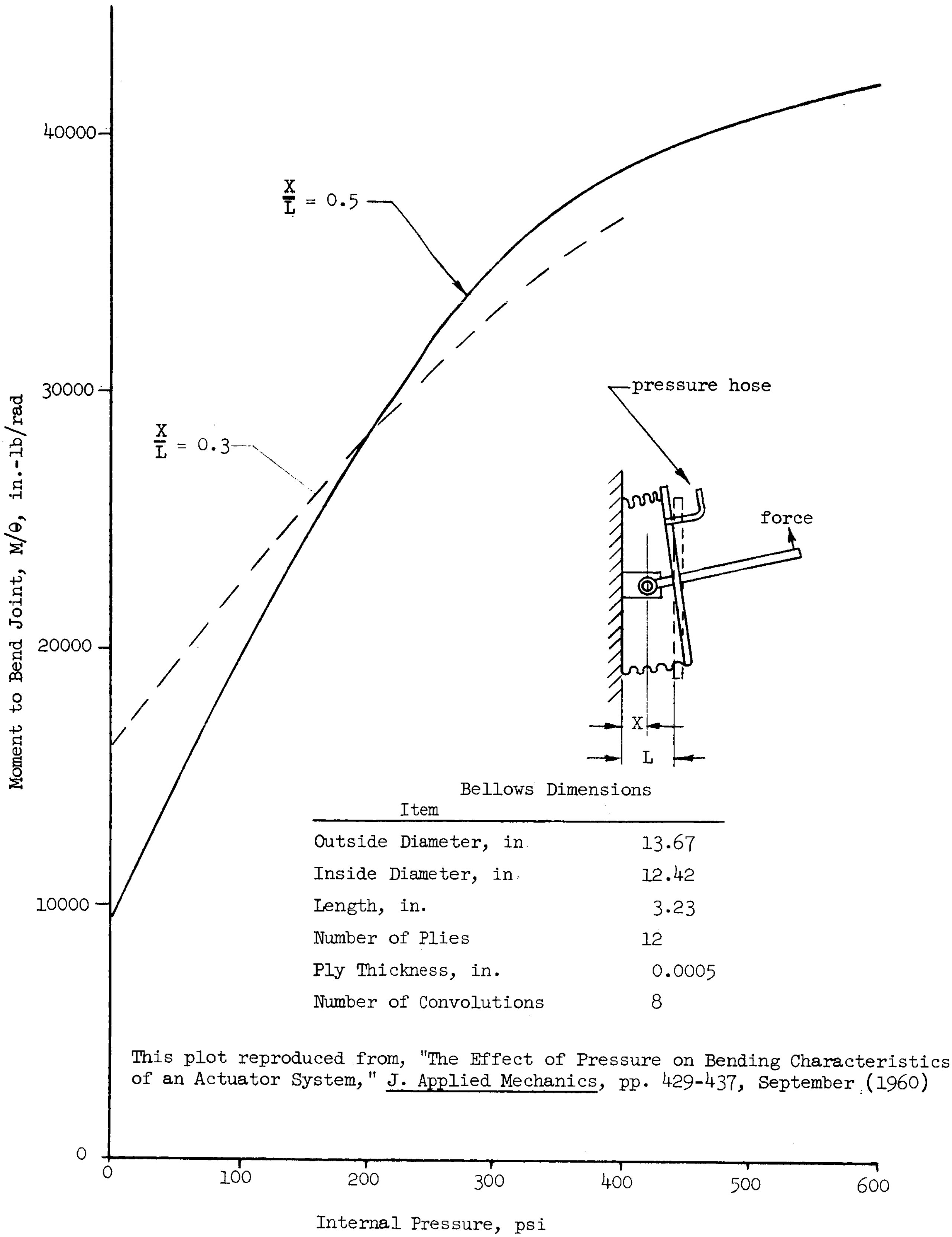


Figure VIII-C-6

3.5-in. 8400 Proof Bellows Joint (Interior)



Bellows Dimensions

Item	
Outside Diameter, in	13.67
Inside Diameter, in	12.42
Length, in.	3.23
Number of Plies	12
Ply Thickness, in.	0.0005
Number of Convolutions	8

This plot reproduced from, "The Effect of Pressure on Bending Characteristics of an Actuator System," J. Applied Mechanics, pp. 429-437, September (1960)

Effect of Internal Pressure on Moment Required to Bend Hinged Joint

Figure VIII-C-7

VIII, Lines, Ducts, and Flexible Joints (cont.)

D. BELLOWS JOINT DESIGN PROBLEMS AND REQUIRED TECHNOLOGY

The design of a satisfactory gimbaling bellows-type joint is dependent on the design of both the bellows element and the restraint device. The two design problems are essentially independent, however, and will be treated separately.

1. Bellows

The flexible joint-design requirements for high-pressure launch-vehicle rocket engines are far more severe than those for engines presently under development. Bellows up to 20-in. dia and 10,000-psi proof pressure are considered in high-pressure engine designs as compared with 10-in. dia and 2500-psi proof pressure for current engines. The bellows being developed for present-day engines are designed largely on the basis of extrapolated empirical data and represent the limitations of the present technology.

A review of the engine-design studies indicates that engines of higher thrust level and pressure tend to require transfer lines of increasingly smaller length-to-diameter ratios. This trend not only reduces the inherent flexibility of the line, but provides proportionately less length to accommodate flexible joints. This trend also demands larger gimbaling or bending capabilities of these joints. To further complicate the bellows-design problem, the high pressure and relatively high velocity of the fluid may require such increases in the cross-sectional area of internal-restraint devices that the bellows ID must be increased to maintain reasonable fluid pressure drop.

The anticipated size and pressure requirements for these large, high-pressure engines are indicated graphically in Figure VIII-C-1. The summary of lines, Table VIII-A-1, gives representative nominal line diameters as well as fluid pressures and temperatures for a variety of engine lines, any of which might require bellows joints.

VIII, D, Bellows Joint Design Problems and Required Technology (cont.)

Because only formed bellows normally lend themselves to transfer-line flexible joint application, the significant advances which have been made recently in the analytical design techniques for welded bellows, especially relative to positive-expulsion device applications,[(1)] are of relatively little value in the analysis of high-pressure formed bellows.

Of particular significance is the fact that very little in the way of analytical design procedures for formed bellows was found in the literature. Much of the information that was reviewed[(2-5)] was found to be applicable only to single-ply bellows and, therefore, is not particularly applicable to high-pressure service. The primary difficulty in establishing what is actually known about the design of high-pressure bellows throughout the industry stemmed from the fact that all bellows manufacturers contacted consider their design procedures to be proprietary. Most indications point to the use of empirical data for practically all design work. This means that presently the design of bellows of the sort needed for large, high-pressure engines would be based on extrapolation of the empirical data currently available.

The present state of the art in the production of formed bellows joints, relative to size and pressure, appears to be more a function of the demand to date rather than any limits of feasibility. The general consensus of opinion among those with whom these limits have been discussed is that bellows joints for the large

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(1) Study of Zero-Gravity Positive Expulsion Techniques, Bell Aerosystems, Report 8230-933004, June 1963.
(2) C. M. Daniels, "Designing for Duct Flexibility with Bellows Joints," Machine Design, 147-155, October 15, 1959.
(3) A. Samoiloff, "Evaluation of Expansion - Joint Behavior," Design and Equipment Application, Section 57-69.
(4) P. Seide, "The Effect of Pressure on the Bending Characteristics of an Actuator System," Journal of Applied Mechanics, 429-437, September 1960.
(5) J. D. Matheny, "Bellows Spring Rate for Seven Typical Convolution Shapes," Machine Design, 137-139, January 4, 1962.

sizes and high pressures could probably be developed, but at relatively large cost in terms of time and money. In the case of most vendors, additional facilities would be required to produce and test these bellows.

If the demand existed at present for large, high-pressure bellows, it is reasonable to assume that these units could eventually be satisfactorily produced by extrapolation of empirical design data followed by trial and error hardware development.

The considerations discussed so far have been limited to the gross functional parameters of size and pressure. The comprehensive analysis of bellows design must of necessity consider other highly important factors, such as number of convolutions, convolution height, convolution pitch, convolution configuration (straight sided, omega, circular, etc.), spring rate, free length, material, temperatures, number of plies, stability requirements, cycle life, mode of operation, and method of forming. These factors cannot be treated independently, but must be considered integral parts of the problem.

At least one bellows manufacturer has indicated that analytical techniques are now available that did not exist four years ago for the design of high-pressure bellows. These techniques would permit the design of bellows with diameters up to 20 in. and proof pressures of 8000 to 13,000 psi as required for advanced-engine designs. This same manufacturer has indicated that the fabrication of such bellows to the design specification would present far more difficult problems than the analytical design. The proprietary nature of this design procedure has precluded evaluation in the course of this program, as well as any possibility of presenting the procedure in this report.

A need exists, therefore, to verify the feasibility of designing and fabricating large-diameter bellows for high-pressure service. Such verification

VIII, D, Bellows Joint Design Problems and Required Technology (cont.)

will necessarily entail full-scale experimentation with emphasis on obtaining low, accurately-predictable spring rate for gimbaling up to $\pm 12°$ while retaining bellows stability. Such a program is outlined in Section VIII, E.

2. Restraint Devices

The solution of all problems associated with the bellows design does not necessarily result in efficient designs for high-pressure flexible joints if consideration is not given to the improvement of the axial-restraint device. With the exception of bellows joints designed expressly for axial motion or for single bellows in extremely short lines, practically all rocket-engine flexible joints require a pivoting axial-restraint device.

The design of axial-restraint devices for high-pressure flexible joints increases in difficulty at a rate directly proportional to the increase in pressure.

For example, a restraint device for a 20-in.-dia bellows designed for a proof pressure 8000 psi must be capable of gimbaling freely under an axial load of over 2.5×10^6 lb.

To establish design criteria for axial restraints which will permit optimum joint design and minimize the constraints placed on the location of the joint in the line system, the following questions must be answered:

(1) What design configuration provides the maximum load-carrying capacity for an internal restraint device while providing minimum blockage to free stream-flow area?

(2) What materials are best suited for unlubricated bearing surfaces in such a design for temperatures ranging from -423 to +1500°F?

VIII, D, Bellows Joint Design Proglems and Required Technology (cont.)

(3) What are the firm criteria for determining when external restraint should be used in lieu of, or in addition to, an internal-restraint device?

(4) What constitutes optimum design of such an external restraint?

(5) What contribution does the friction of a well-designed restraint device make towards the total moment required to gimbal the joint?

VIII, Lines, Ducts and Flexible Joints (cont.)

E. RECOMMENDED ADVANCED TECHNOLOGY BELLOWS JOINT PROGRAM

In order that the feasibility of producing bellows-type flexible joints to meet the requirements of large, high-pressure rocket engines may be verified, a demonstration of design and fabrication technology is required both for the bellows proper, and for the axial-restraint device where this is incorporated. The design and fabrication techniques used in this verification should be made available for determining the feasibility of manufacturing hardware for specific engine requirements. The program results should be applicable to formed metal bellows for flexible joints and ducts from 5- to 20-in. diameter with internal operating pressures from 100 to 6600 psi (120 to 13,000 psi proof, depending on the factor of safety), for fluid media including liquid hydrogen, liquid oxygen, and hydrocarbon propellants, as well as hot-gas products of combustion at temperatures up to 1500°F.

To simplify the program, it is recommended that demonstration be limited to a 20-in.-dia, 10,000-psi proof (6600 x 1.5 F.S.), cryogenic bellows joint and a similar 8000-psi (0.80 x 6600 x 1.5 F.S.) hot-gas joint. The cryogenic unit would be designed for a temperature range from -423°F to 100°F with emphasis on the spring rate at low temperature, and the hot-gas unit would be designed for 0 to 1500°F with emphasis on strength and stability at high temperature.

Fluid velocities would be considered up to the following limits of nominal line velocity for purposes of analysis.

LH_2,	1000 ft/sec
LO_2,	500 ft/sec
Hydrocarbon,	200 ft/sec
Hot Gas,	0.3 Mach

Any actual flow testing required to evaluate pressure drop may be carried out with water.

VIII, E, Recommended Advanced Technology Bellows Joint Program (cont.)

Cycle life should be at least 500 cycles

Gimbaling or bending requirements for both bellows should be treated as follows:

Primary Goal (Desired capability) $\pm$ 12°

Secondary Goal (Acceptable capability) $\pm$ 7°

Minimum Requirement (Requiring installation at full deflection) $\pm$ 3-1/2° or + 7° - 0°

Lateral displacement (shear deformation) and axial-translation (expansion and compression) limitations may be treated analytically with verification established by testing the same bellows hardware designed for the hinged joint, rather than designing separate units for particular specifications.

The results of this program should include the following:

1. Presentation of an analytical method of predicting the relationships between deflection, stress, and stability of bellows for those conditions encountered in flexible-joint application. These conditions include simple axisymmetric expansion and contraction as well as nonaxisymmetric gimbaling and lateral deflection. The feasibility of analyzing simultaneous combinations of these conditions should also be evaluated. Unrestrained as well as restrained operation should be considered.

Essential to this analysis will be a means of describing the bellows structure, taking into account the following descriptive parameters:

Internal Diameter

Convolution Configuration (precise description)

VIII, E, Recommended Advanced Technology Bellows Joint Program (cont.)

Length (free and as-installed)

Number of convolutions

Ply thickness (as a function of position on a given convolution, if variation due to forming is significant)

Number of Plies

Material Properties (Including postforming non-linearities due to cold working, etc.)

This analysis should also indicate whether the applications and designs under consideration require sophisticated vibration analysis similar to that required for welded bellows for positive expulsion devices.* This phase of the study should take into account fluid-dynamic induced forcing functions associated with high-velocity flow, especially where the bellows is located immediately downstream of an elbow.

2. Explanation of the techniques for the design or performance prediction of bellows for any application within the scope of size, pressure, and temperature set forth above. These techniques would logically be expected to be capable of establishing limits to the applicable functional parameters, such as the maximum pressure for which a given diameter bellows might be designed without exceeding a particular spring rate.

3. Evaluation of results of the experimental program which:

a. Verify the analytical method of predicting stress, deflection, and stability relationships.

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* Study of Zero-Gravity Positive Expulsion Techniques, Bell Aerosystems No. 8230-933004, June 1963.

VIII, E, Recommended Advanced Technology Bellows Joint Program (cont.)

b. Evaluate the accuracy of the design techniques.

c. Indicate those fabrication problem areas resulting in finished hardware that does not coincide precisely with design drawings. Where such deviations from specification cannot be remedied with confidence, means should be established for predicting these deviations and modifying the design procedure to compensate.

4. Evaluation of design concepts for axial-restraint devices (internal, external, or combined) to establish design criteria for high-pressure, high fluid-velocity application. Primary emphasis should be on:

a. Effect on bellows-design requirements (i.e., constraints imposed on type of motion; locally increased fluid velocity; elimination of torsional loads on bellows).

b. Pressure drop.

c. Effect of restraint friction on overall joint flexibility or spring rate.

d. Limitations of particular concepts due to the type of motion that may be accomodated.

Consideration must also be given to:

a. Weight

b. Envelope size

c. Reliability

d. Cost

VIII, E, Recommended Advanced Technology Bellows Joint Program (cont.)

Design criteria, considering the full range of functional applications, should be set forth for restraint devices for each of the following modes of motion:

a. Single-plane bending (hinging).

b. Any-plane bending (gimbaling).

c. Single-plane bending, plus lateral displacement.

d. Any-plane bending, plus lateral displacement.

5. Criteria for the design of flexible joints for any specified application within the scope of the functional requirements previously outlined. Such joint-design criteria must have provision of treatment of:

a. Minimum joint spring rate commensurate with functional constraints and production feasibility or practicality.

b. Bellows-liner requirement due to pressure drop or dynamic-stability considerations.

c. Accurate performance prediction in terms of pressure drop, joint spring rate, weight, cycle life, and response frequency.

d. Optimum materials selection.

e. Ability to transmit or absorb torsional loading, and effect of such loading on joint performance.

VIII, E, Recommended Advanced Technology Bellows Joint Program (cont.)

The results of this program should be so presented as to provide a guide for the design of actual engine-system flexible joints, as well as provide a ready means whereby the duct end reactions and duct stresses may be accurately predicted for those systems to which the joints are applied. These results should show, for example, how such functional parameters as spring rate, cycle life and stability are affected by varying the number of plies while holding constant all other descriptive parameters such as total wall thickness, number of convolutions, etc. The determination of those parameters to be varied during the experimental phase of the program must of necessity be based upon results of the analytical investigation. The large number of descriptive parameters and the high cost of fabricating and testing different bellows designs will require careful selection of variables and maximum use of statistical test planning to obtain sufficient experimental data to verify the analytical results.

F. FILAMENT-WOUND REINFORCED PLASTIC LINES

In Section VIII, B, it was pointed out that one of the three basic means of obtaining greater line flexibility is to vary the material properties of the line. Filament-wound, reinforced-plastic lines and ducts offer sufficient potential advantages to warrant further evaluation of the technology involved. The parametric study of line-material properties, Appendix G, showed the desirability of high values of $\frac{S}{E}$ for flexibility and $\frac{S^2}{\rho E}$ for combined flexibility and weight considerations. Filament winding of various materials appears to be an excellent means of tailoring allowable stress and elastic modulus directionally to optimize the desired properties for particular applications. This section will discuss the potential advantages of this technique as well as certain limitations and development problems.

1. Advantages

Filament-wound lines offer potential advantages over conventional lines in the areas of flexibility, weight, heat transfer and fabricability. A comparison of the properties of filament-wound material with those of some metals is shown in Figure VIII-F-1. The filament-wound composite possesses high strength, low density, and low modulus when compared with the other materials shown. The tests summarized in Table VIII-F-1 indicate the bending properties of unpressurized filament-wound tubes.

a. Flexibility and Weight

The parametric study of material properties affecting line flexibility and weight indicated that the most important parameters are allowable stress in the hoop direction, elastic modulus in the axial direction, and density of the material. The initial effort consisted of evaluating materials of construction whose properties would yield a maximum value of flexibility factor, $f \sim \frac{S}{E}$, or performance factor, $PF \sim \frac{S^2}{\rho E}$. Properties of some representative metals are shown in Table VIII-F-2. Table VIII-F-3 includes rough cost estimates and some cryogenic property data for seven metals. It was considered appropriate to establish some goal or desired properties for flexible, light-weight ducting. The values selected were a yield stress of 150 x 10^3 psi and a modulus and density of 0.2 x 10^6 psi and 0.05 lb/in.^{3}, respectively. These resulted in values of $\frac{S}{E}$ = 500 x 10^{-3} and $\frac{S^2}{\rho E}$ = 1000 x 10^3 (Figure VIII-F-1).

In view of the apparent potential of filament-wound plastics, serious consideration was given to the feasibility of varying material properties directionally. Because a cylindrical line under pressure only sees an axial stress of one-half the hoop-stress value, a material with an allowable hoop stress twice its longitudinal stress would be more efficient than a homogeneous material.

The addition of bending loads requires an increase in the axial stress, but this may be minimized by use of a low-modulus material in the axial direction. To control volumetric change with pressure, the modulus in the hoop direction should be relatively high. Therefore, the optimum line material will have high strength and modulus in the hoop direction and somewhat lower strength and much lower modulus in the axial direction. This applies to relatively high-pressure lines. Low-pressure applications with relatively large bending loads could require higher allowable stress in the axial direction than in the hoop direction.

Filament-wound reinforced-plastic construction provides the capability of varying properties directionally, either by varying the winding pattern or by using filaments of different materials for the hoop and axial directions. Considering S to be hoop stress and E to be axial modulus, a conventional filament-wound fiber-glass line would have an $\frac{S}{E}$ of 81.4 x 10^{-3} and an $\frac{S^2}{\rho E}$ of 159.6 x 10^3. By comparison, a proposed construction, using very low-modulus reinforcement (i.e., Nylon, Dacron, etc.) in the axial direction, would raise these factors to 500 x 10^{-3} for $\frac{S}{E}$ and 1000 x 10^3 for $\frac{S^2}{\rho E}$, which correspond with the "ultimate" values established as goals for flexible, high-pressure lines.

VIII, F, Filament-Wound Reinforced Plastic Lines (cont.)

Complex line geometries may require that directional material properties (or even the cross-sectional shape) be varied along the length of a given line to accommodate torsional loads or large differences in bending loads, while at the same time providing maximum flexibility. Conventional metal lines usually have a circular cross section and uniform wall thickness over the entire length of the line. The thickness of these metal lines is determined by the most severe stress conditions anywhere in the line. With certain geometries, an increase in flexibility in a particular plane may be obtained by the use of elliptical or other noncircular cross sections in bends. To minimize fluid pressure drop, it is desirable to maintain a constant cross-sectional area, which in turn requires a larger periphery on the inside wall for the noncircular section. The following discussion includes the feasibility of fabricating such variable sections.

b. Fabrication Feasibility

Filament-wound construction allows variation of the directional properties to suit the loading conditions. For example, an isotensoid pattern (equal tensile loading in all filaments) may be achieved in a straight duct subjected to internal pressure by placing longitudinal and hoop windings in a 1:2 ratio or by winding helically at a 54.75° angle to the axis.* If the pressure is increased, the duct strength can be increased by merely adding more material in the same pattern, with no additional tooling required. When additional loads such as bending, axial compression, axial tension and/or torsion are introduced, the winding angles and/or the ratio of thicknesses of the windings may be changed to suit these conditions. The directional properties can be varied along the length of the line. Cross-sectional shape and/or diameter may also be varied along the length of the line to allow changes in section modulus or to reduce pressure drop.

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* F. J. Darms, R. Molho, "Optimum Filament-Wound Construction for Cylindrical Pressure Vessels," Aerojet-General Corporation, August 1962.

VIII, F, Filament-Wound Reinforced Plastic Lines (cont.)

In a structural analysis performed on a filament-wound toroidal pressure vessel, the cross section and winding pattern were determined, which allowed isotensoid loading in the filaments. Aerojet has tested a specimen (Figure VIII-F-2) that was built using this analysis as a basis for design. Burst pressure was 8300 psig with a filament stress of 310,000 psi. The cross section was noncircular and a continuously varying helix angle was employed. Figure VIII-F-3 shows the toroid winding machine. The success of this initial test indicates that analytical methods can be successfully derived for nonlinear, tubular, pressurized structures and that technology for fabricating such vessels can also be readily developed.

Figure VIII-F-4 shows a winding machine developed at Rohr Corporation, Riverside, California for winding low-pressure thin-walled hot-air ducts for commercial aircraft.* In the photograph the machine is winding on an S shaped mandrel. While no provisions are made for continuously varying the helix angle or for curvature in the vertical plane, it is evident that this basic principle could be extended to allow the winding of almost any pattern, size, or shape.

c. Properties of Presently Available Materials

Present state-of-the-art materials for filament-wound rocket cases are S glass filaments and epoxy resins. The filament-winding process can be used with any number of other reinforcements and matrix materials. For example, Dacron yarn, while retaining a respectable strength-to-density ratio in a pressurized tubular structure of approximately 1×10^6 in., would be given an apparent longitudinal modulus of roughly 0.2 by 10^6 psi. Figure VIII-F-5 shows a typical stress-strain curve for Dacron yarn. Matrix materials can be chosen to suit environmental and/or fabrication conditions. There are matrix resins that would extend the upper temperature limit to roughly 800°F.** Recent Aerojet Polaris investigations have shown that

* Boeing Company Material Specification BMS 8-57 "Glass Fiber-Reinforced Fire-Resistant Plastic Air Ducts."

** "Polybenzimidazole Resins for High-Temperature Reinforced Plastics and Adhesives," ASD Contract AF 33(657)-8047, Narmco Research and Development, Division of Telecomputing Corporation.

VIII, F, Filament-Wound Reinforced Plastic Lines (cont.)

filament-wound pressure vessels have low notch sensitivity and are amenable to field repairs.

Little data is presently available on thermal properties of filament-wound reinforced-plastic materials at cryogenic temperatures, but the following values* for epoxy glass-cloth laminate are representative:

Temperature t (°F)	Coefficient of Thermal Expansion α (in./in./°F)	Coefficient of Thermal Conductivity K (Btu/hr/ft^2/in./°F)	Specific Heat c_p (Btu/lb°F)
-100	7×10^{-6}	0.9	0.301
+100	7×10^{-6}	1.15	0.301
+400	2×10^{-6}	1.4	0.301

In addition, Table VIII-F-4 shows the estimated effect of cryogenic temperatures on the strength and modulus of a filament-wound structure, based upon available glass laminate data.

Filament-wound structures possess other characteristics that are beneficial for transfer-line applications, such as high vibration-damping capability, high electrical resistivity, transparency to electromagnetic radiation, and corrosion resistance.

2. Problems and Potential Solutions

a. Liners

Certain problems will have to be overcome to develop practical filament-wound transfer lines. One problem is the inherently high porosity of the

* Measurement of the Thermal Properties of Various Aircraft Structural Materials, Wright Air Development Center, Report 57-10.

VIII, F, Filament-Wound Reinforced Plastic Lines (cont.)

material. The Boeing Company specification for hot-air ducts* allows as much as $14{,}200 \dfrac{(\text{Std in.}^3)\ (\text{mil})}{(24\ \text{hr})\ (100\ \text{in.}^2)\ (\text{Atmosphere})}$ air leakage. This would be excessive for rocket-engine application. Low-pressure filament-wound chemical pipe lines operate successfully without liners by using high resin contents and low stress levels; however, most high-efficiency filament-wound pressurized structures utilize elastomeric liners. Metal liners, currently under development, can be used and offer lowest permeability, but thermal and mechanical-strain compatibility requires patterning of the metal to allow for the high elongations of the fiber-glass structure. Cryogenic usage may preclude the use of any but a small group of organic polymers, not only because of the requirements for good physical properties at low temperatures but also because of impact sensitivity with liquid oxygen.

The fluorocarbon polymers are the most likely candidates for service with liquid oxygen and liquid hydrogen. The fluorocarbons listed below are inherently resistant to the propellants under consideration:

(1) Bonding (Fusing) Fluorocarbon Sheet Stock

Another and probably the most obvious method of liner fabrication is that of either longitudinal or circumferential fusing (bonding) fluorocarbon sheet or film. Of these two methods, the longitudinal process is probably the most suitable, because this reduces the number of fused joints and possibility of weak porous areas in the part.

(2) Metallic-Fluorocarbon Composite Liner

Several companies have had a certain degree of success with the fabrication of metal-fluorocarbon laminates. This concept looks promising

* op. cit.

because of the certainty of having zero-permeability through the liner due to the metallic portion of the laminate.

b. Flammability

The resins normally used in filament winding are flammable, however, additives are available that will prevent the resin from supporting combustion without adversely affecting its other properties. The Boeing hot-air ducts, for example, must pass rigid flame-resistance tests. These additives would not prevent charring from flash fires caused by propellant leakage, but the low thermal conductivity will minimize char penetration. A thin "sacrificial" layer of extra windings or insulation can probably be developed to control charring, and field repairs will allow correction of minor damage.

c. Temperature Limitations

The 500°F service required of aircraft hot-air ducts is the highest known sustained operating temperature for present state-of-the-art-materials. However, several resins are under development that promise to increase the useful temperature limit for sustained service to approximately 800°F.* For short-term exposure, insulation or ablative liners can increase practical operating temperatures. Figure VIII-F-6 shows the results of some Aerojet work on the Polaris program, indicating a 60% longitudinal filament-strength retention at 500°F steady-state temperature.

d. Properties at Cryogenic Temperatures

As indicated previously, the cryogenic performance of filament-wound structures is not completely known; however, extrapolations based on more complete

* "Polybenzimidazole Resins for High-Temperature Reinforced Plastics and Adhesives," ASD Contract AF 33(657)-8047, Narmco Research and Development Division of Telecomputing Corporation.

VIII, F, Filament-Wound Reinforced Plastic Lines (cont.)

data available for glass-cloth laminates (which contain the same basic materials) indicate that improved properties are expected compared with those at room temperature (Table VIII-F-4).

Another area of meager data concerns the effects of sustained loading or cyclic loading at cryogenic temperatures on filament-wound structures. NASA Contract NAS 3-2562 promises to fill in some of these blanks; this data will be directly applicable to filament-wound lines.

e. Connectors

Joining of line sections will entail all the same problems of joint and seal design encountered with metallic lines. No reference has been found in the literature on connectors designed for use at pressures of 6000 psi. Designs mentioned in the literature are mostly for lower pressure (up to 1200 psi) and non-cryogenic service. Such joints are used in aircraft hot-air ducts. For removable joints, the filament-wound line may have to contain metal end fittings. One method is the use of a flared pipe end with a split tapered metal flange bonded to the flare. It has been determined that the flared length must be approximately 0.4 times the diameter. Another concept is that of forming threads during the winding operation, threading the propellant line to a flange, and bolting or welding the mating metal flanges together. This method appears feasible as it is possible to fabricate glass-filament-wound parts with the threads wrapped in the part (Figure VIII-F-7). This is much more desirable than machining the threads, which considerably weakens the composite structure. The part shown is a commercial item used on electrical transformers, which are proof-tested at 2700 psi before shipment. They have also been burst-tested to above 4000 psi and failure normally does not occur at the threads. Joint designs used for solid-rocket cases and pipes can probably be adopted for propellant-transfer lines.

TABLE VIII-F-1

RESULTS OF BENDING TESTS ON FILAMENT-WOUND TUBING

Source	Test Conditions	Specimen Configuration	Materials of Construction	Winding Pattern	E_1 (Longitudinal Modulus), x 10^6 psi	S_1 (Ultimate Stress), psi
Lamtex	Room Temp 3-Point Loading Simply Supported	1-in. ID x 18-in. long x 1/16-in. thick (4 tests)	E-801 Yarn with epoxy resin	54.8° Helical	3.11	25,600
Kidde		Same as above but 24- to 25-in. long (4 tests)		54.8° Helical	1.61	12,500
		Same as above but 24- to 32.6-in. long (4 tests)		L-C	1.80	43,700
	Room Temp Cantilever with End Load	3-in. ID x 24-in. long x 0.065-in. thick (one test)		L-C	2.44	49,800
	"	" (one test)		54.8° Helical	2.20	19,800
Hughes	250°F Steady-State Cantilever with Pure Bending Movement at Free End	Not Given (3 tests)	E-Glass 60 end roving for hoop windings. 143 glas. cloth for longitudinal reinforcement-phenolic resin	L-C	2.38	22,800
Hughes	Same as above but 950°F Transient Heating	Not Given (3 tests)	"	L-C	0.977	9,350

TABLE VIII-F-2

PROPERTIES OF POTENTIAL LINE MATERIALS

MATERIAL	HOOP STRESS S_H psi x 10^3	AXIAL MODULUS E_{axial} psi x 10^6	DENSITY ρ lb/in.^{3}	S/E** x 10^{-3}	$S^2/\rho E$*** x 10^3
"X" *	100	0.2	0.05	500	1000
Dacron	50	0.2	0.05	275	300
E-glass	150	2	0.07	80	150
Inconel-718	150	30	0.3	5	3
Al 6061	40	10	0.1	4	1.5
347-SS	40	30	0.3	1.5	0.2

* A hypothetical material whose properties were established as a goal for maximum line flexibility with minimum weight. Theoretically, such a material could be approximated by winding Dacron in the axial direction and E-glass in the hoop direction.

** S/E is the material properties function which determines the flexibility factor (f/L).

*** $S^2/\rho E$ is the material properties function which determines material performance factor, PF, $\frac{(f/L)}{(W/L)}$.

TABLE VIII-F-3

PROPERTIES OF VARIOUS METALS

		Temperature, °F			
		R.T.	-110	-320	-423
Al 6061-T6	F_{ty}, ksi	39	42	47	52
	E, x 10^6 psi	10	10.5	11	
	Cost, $/lb	0.54			
	Density, lb/in.^{3}	0.098			
Ti-6Al-4V	F_{ty}, ksi	125	155	210	250
	E, x 10^6 psi	14	15	17	18
	Cost, $/lb	15			
	Density, lb/in.^{3}	0.160			
Inconel X	F_{ty}, ksi	135	140	145	150
	E, x 10^6 psi	30	31	31	32
	Cost, $/lb	1.45			
	Density, lb/in.^{3}	0.300			
301 Stainless Steel (20% Cold Rolled)	F_{ty}, ksi	160	160	165	180
	E, x 10^6 psi	25	28	28	30
	Cost, $/lb	0.45			
	Density, lb/in.^{3}	0.290			
347 Stainless Steel	F_{ty}, ksi	40	45	50	65
	E, x 10^6 psi	28	28	28	29
	Cost, $/lb	0.79			
	Density, lb/in.^{3}	0.286			
18% Ni Maraging Steel	F_{ty}, ksi	245	270	305	345
	E, x 10^6 psi	26.5			
	Cost, $/lb	1.00			
	Density, lb/in.^{3}	0.29			
Inconel 718	F_{ty}, ksi	155	165	170	185
	E, x 10^6 psi	29.6			
	Cost, $/lb	1.45			
	Density, lb/in.^{3}	0.297			

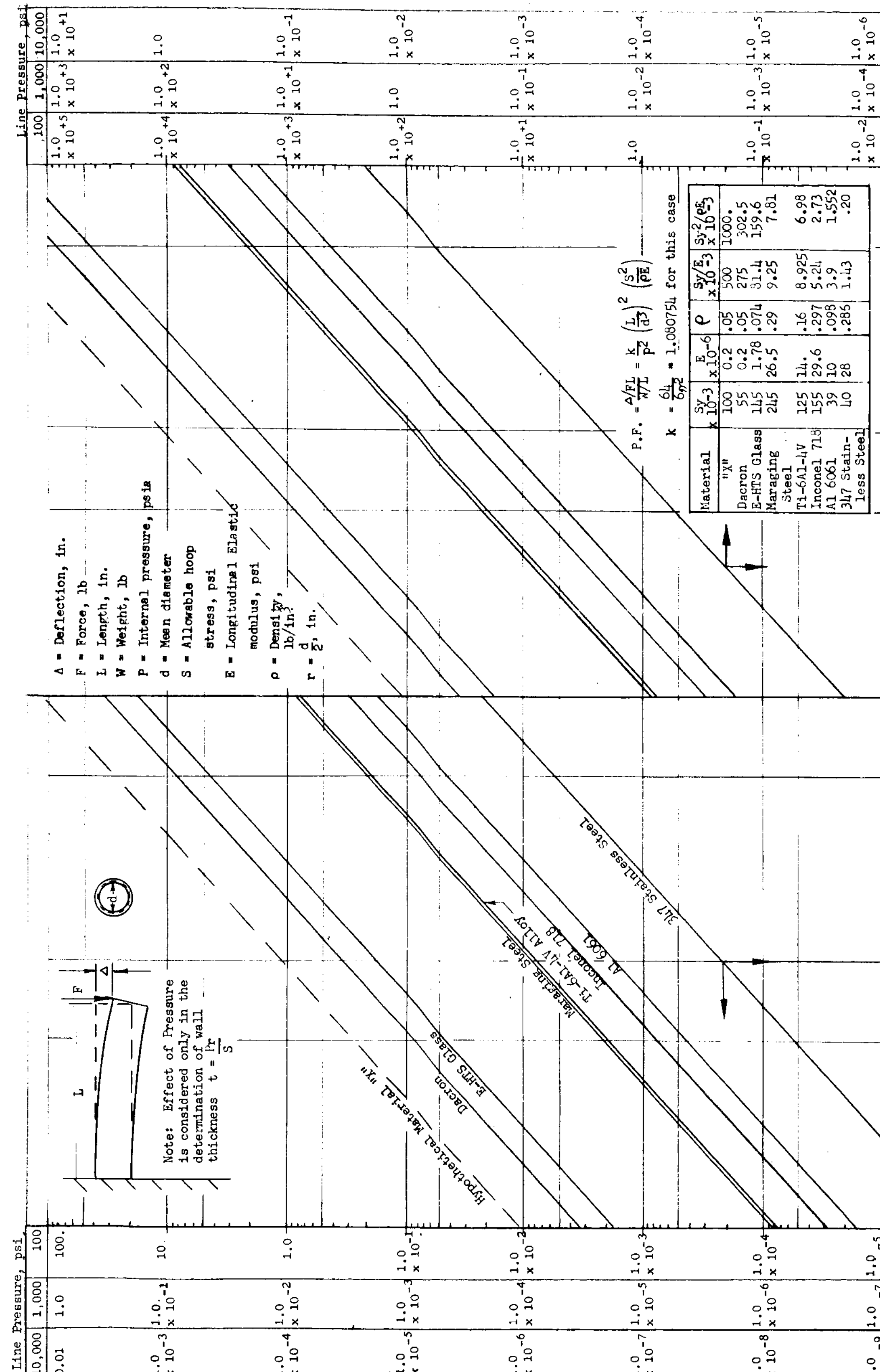
TABLE VIII-F-4

TENSILE STRENGTH AND ELASTIC MODULUS OF GLASS LAMINATE AND FILAMENT COMPOSITE

	Tensile Strength, (psi)		Elastic Modulus, (x 10^6 psi)	
	Glass Laminate*	Filament Composite** (Hoop Direction)	Glass Laminate*	Filament Composite** (Axial Direction)
Room Temperature	40,900 49,000	145,000	3.34 3.51	1.78
-110°F	62,300 71,350	217,500	3.83 3.65	1.95
-320°F	91,600 94,900	304,500	4.15 3.93	2.08
-423°F	97,800 100,700	324,000	4.54 4.26	2.27

* Determination of the performance of plastic laminates under cryogenic temperatures, NARMCO Research and Development. Data shown for two types of epoxy resins.

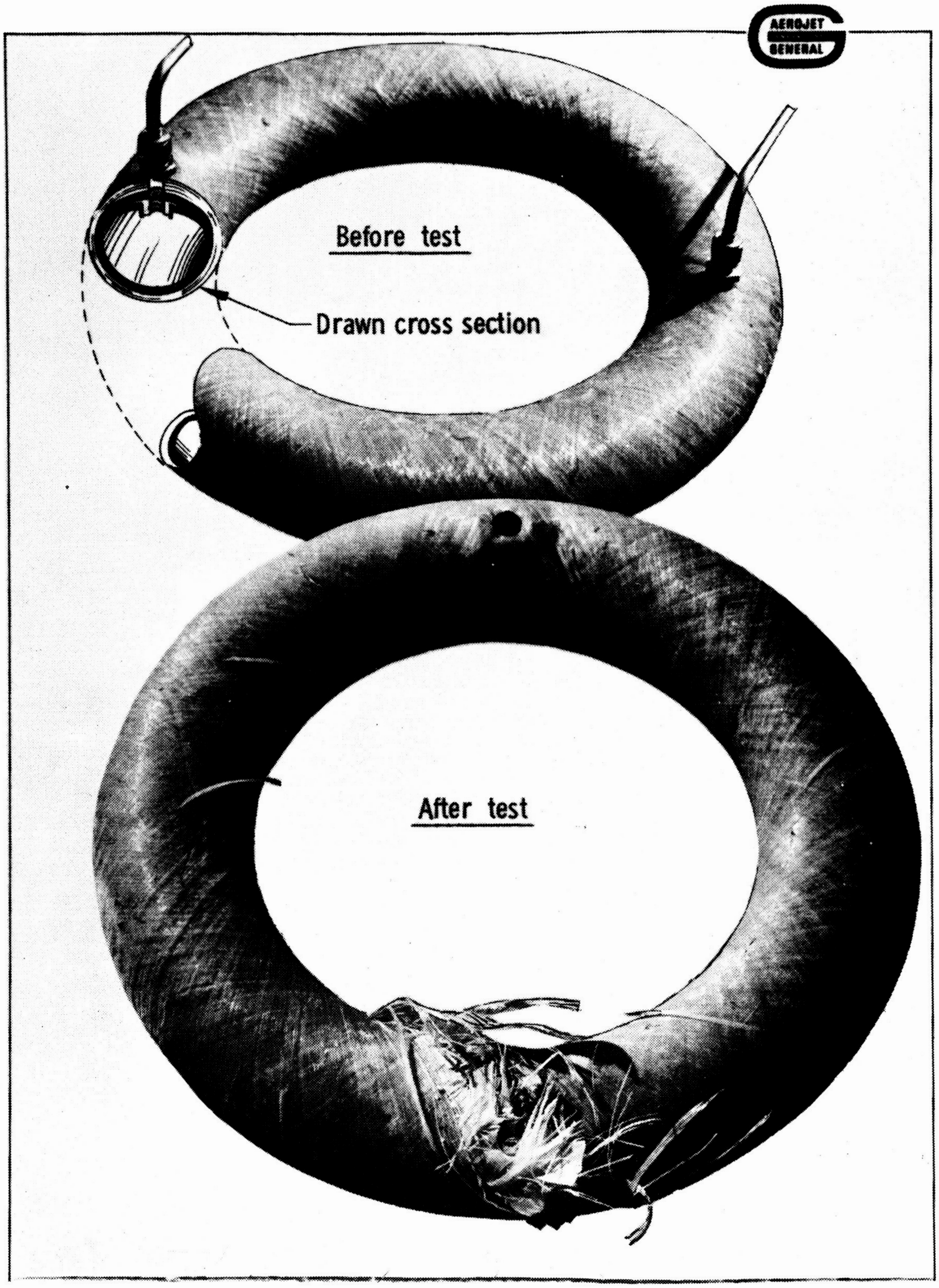
** Estimated values



Material	Sy x 10⁻³	E x 10⁻⁶	ρ	Sy/E x 10⁻³	Sy²/ρE x 10⁻³
"X"	100	0.2	.05	500	1000.
Dacron	55	0.2	.05	275	302.5
E-HTS Glass	145	1.78	.074	81.4	159.6
Maraging Steel	245	26.5	.29	9.25	7.81
Ti-6Al-4V	125	14.	.16	8.925	6.98
Inconel 718	155	29.6	.297	5.24	2.73
Al 6061	39	10	.098	3.9	1.552
347 Stainless Steel	40	28	.285	1.43	.20

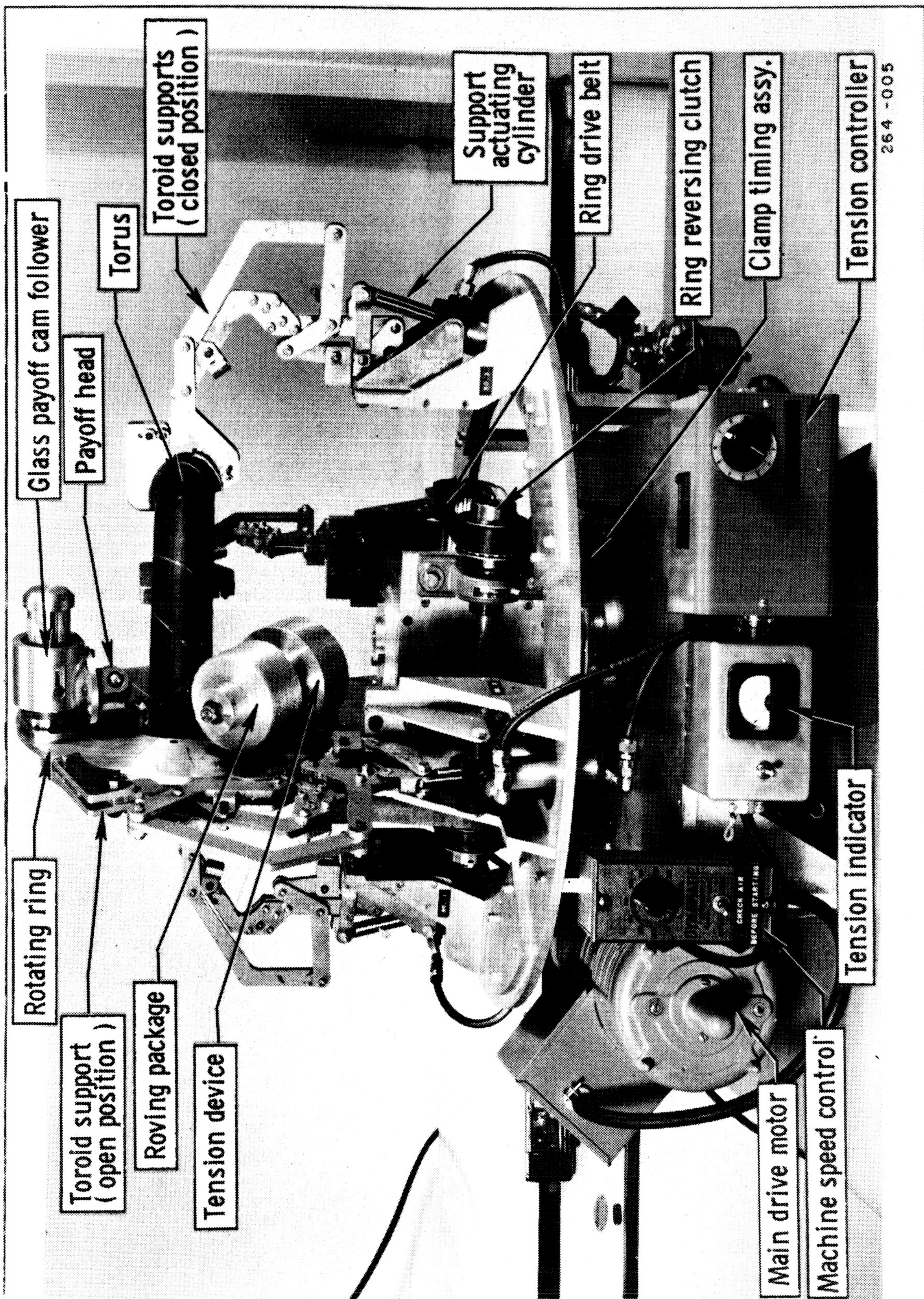
Performance Factor vs $\frac{L}{d^3}$ for Various Metals and Filament-Wound Plastics

Figure VIII-F-1



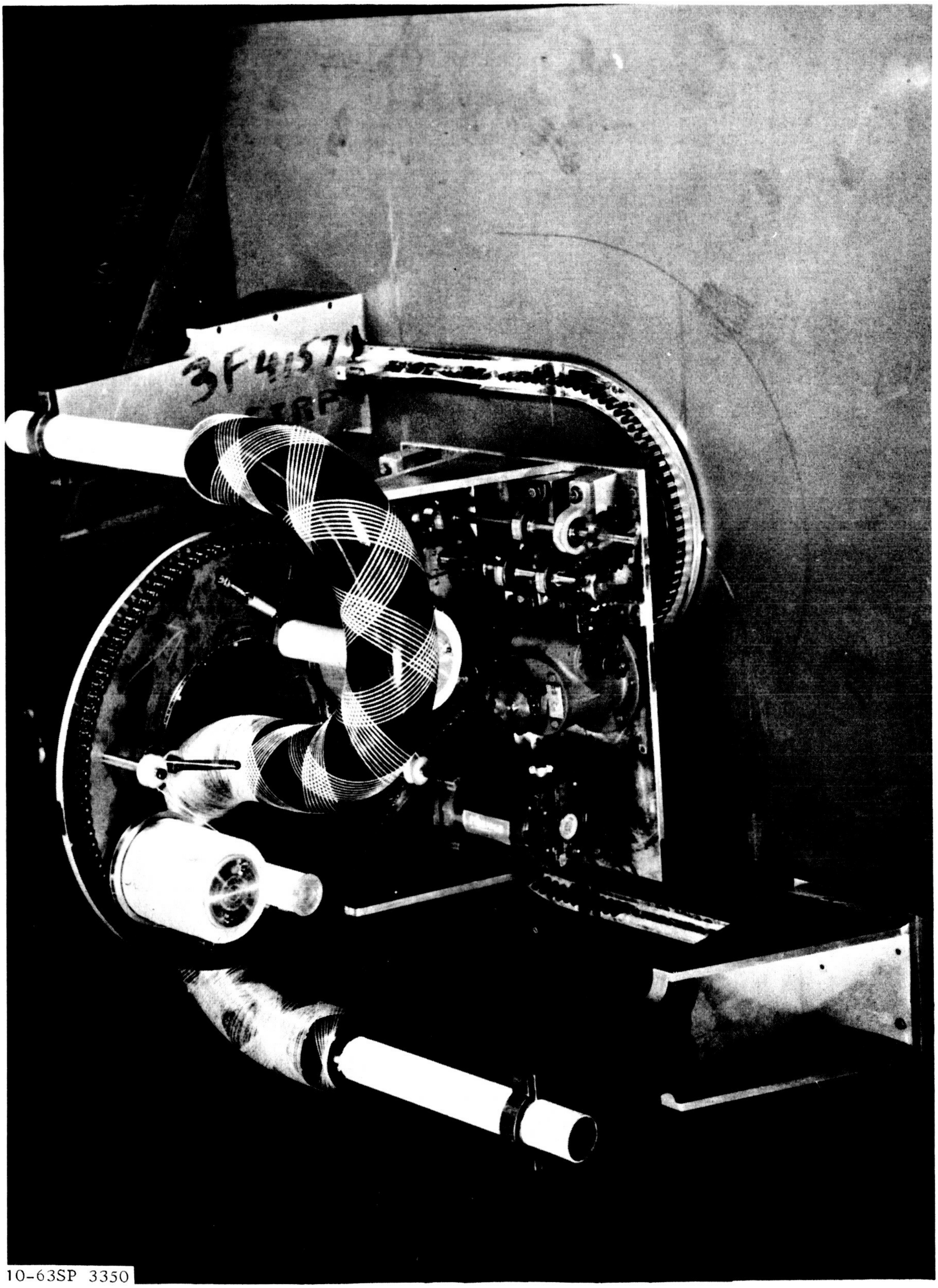
Filament-Wound Geodesic Isotensoid Toroidal Pressure Vessel

Figure VIII-F-2



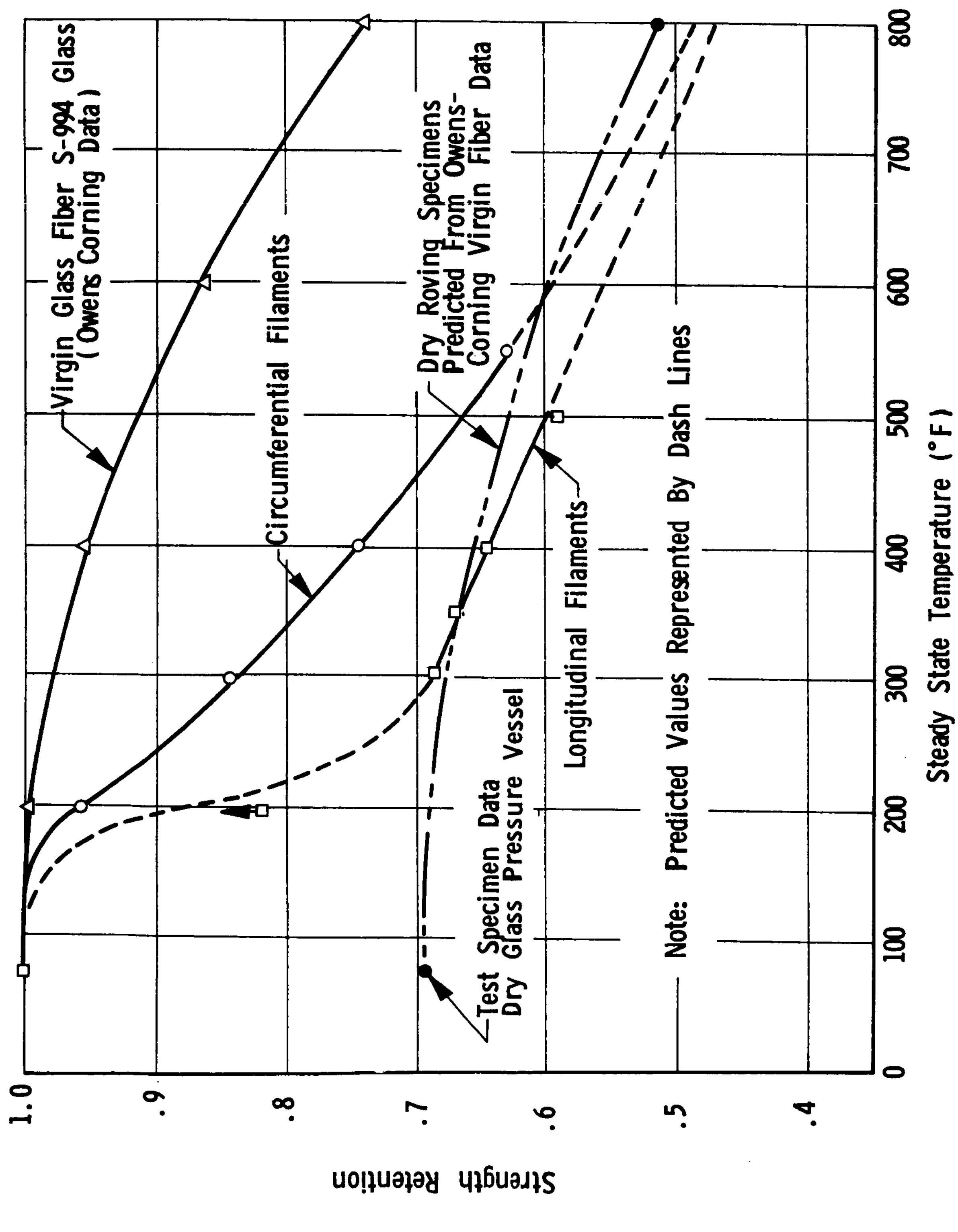
Experimental Toroidal Winder

Figure VIII-F-3



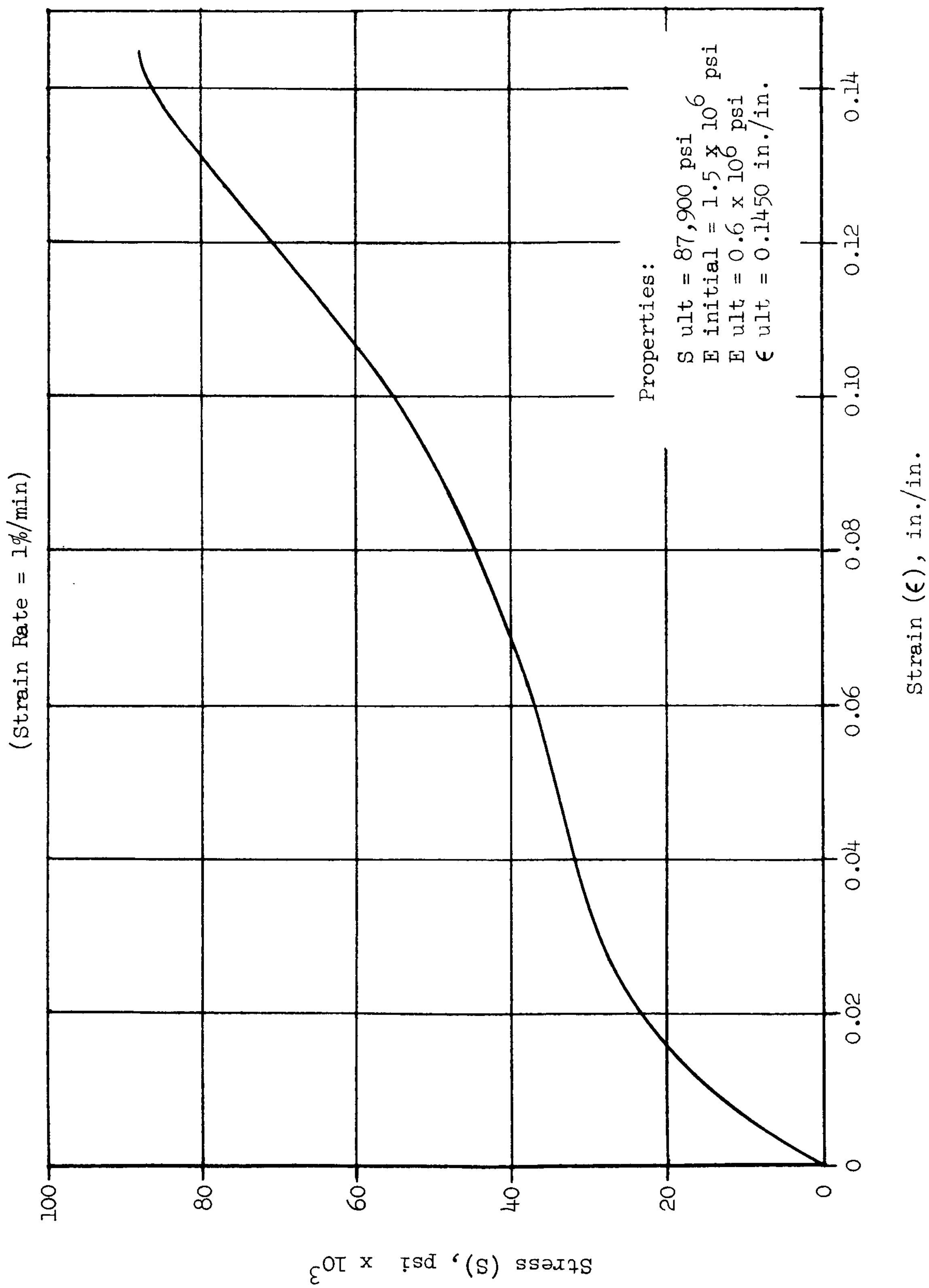
Machine for Winding Complex Ducts and Lines

Figure VIII-F-4



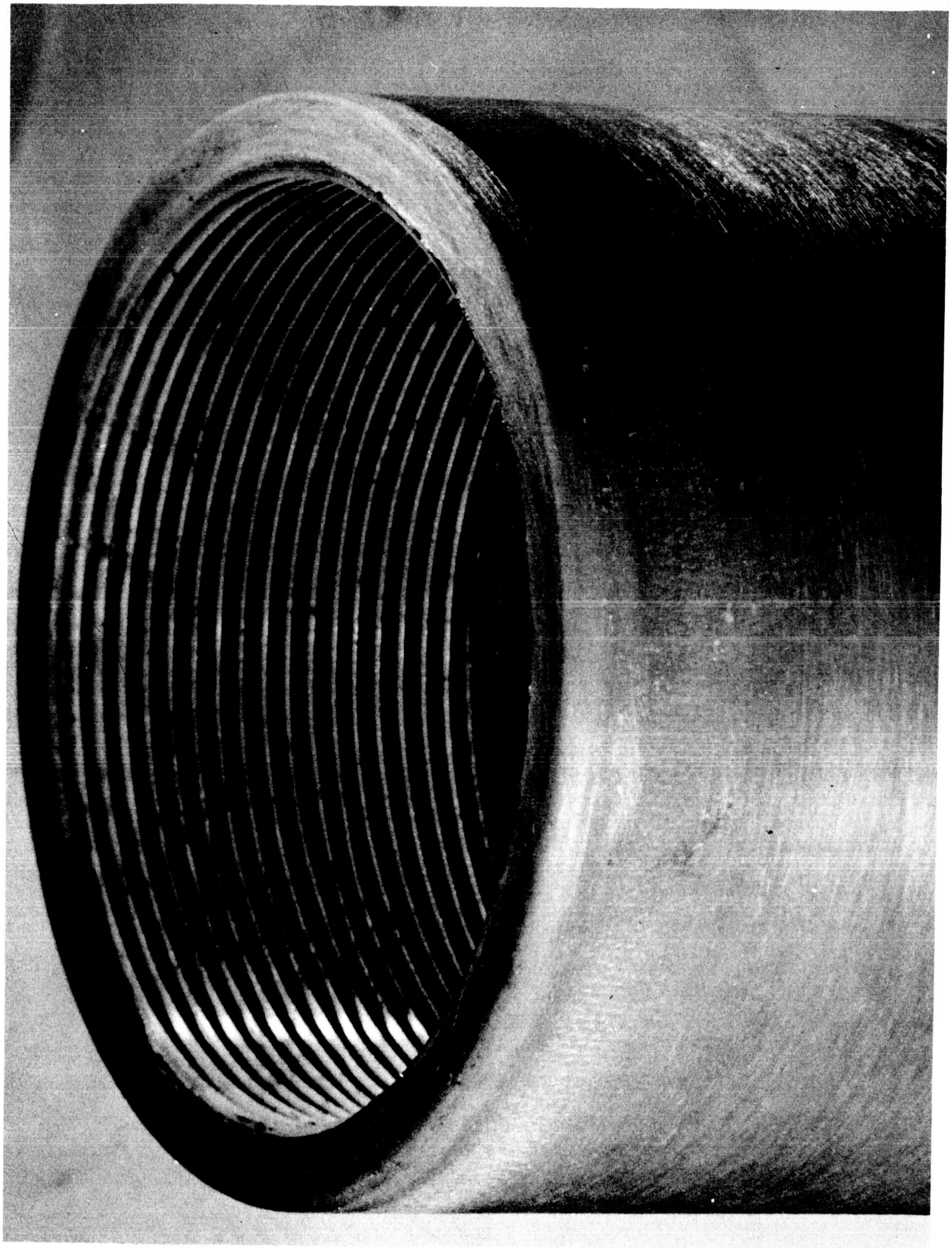
Steady-State Temperature Effects on Strength Retention of Filament-Wound Composites

Figure VIII-F-6



Typical Stress-Strain Curve, 1100 Dacron Yarn

Figure VIII-F-5



Threaded Filament-Wound Transformer Container

Figure VIII-F-7

VIII, Lines, Ducts and Flexible Joints (cont.)

G. FILAMENT-WOUND LINE PROBLEMS AND REQUIRED TECHNOLOGY

The feasibility of using filament-wound reinforced-plastic lines or ducts for large-diameter, high-pressure applications on liquid-rocket engines remains to be demonstrated. Until such feasibility is established, the potential problem areas cannot be considered critical. After the basic techniques for the design and fabrication of high-pressure filament-wound lines are demonstrated by a program such as that recommended in Section VIII, H, it may be expected that advanced-technology work will be required in the following areas:

1. Development of liners, with consideration given to fabrication techniques and means of ensuring adhesion of the liner to the filament-wound structure.

2. Development of connectors for various types of attachment to mating components (i.e., bolted, welded, swaged, etc.).

3. Adaptation of existing computer techniques for the analysis of piping systems and the design of toroidal pressure vessels to the more complex filament-wound line with directionally variable material properties and nonuniform wall thickness at bends.

4. Development of winding equipment with sufficient flexibility to wind a large range of line diameters and configurations, based on existing toroid, pipe, and duct-winding equipment.

5. Further investigation of material properties, especially at cryogenic temperatures, to provide more complete data for structural analysis. Other properties which require better delineation are vibrational damping factor and permeability of the filament-wound composite at cryogenic temperatures.

VIII, Lines, Ducts and Flexible Joints (cont.)

H. RECOMMENDED FILAMENT-WOUND LINE FEASIBILITY DEMONSTRATION PROGRAM

The purpose of the recommended program is to demonstrate the feasibility of designing and fabricating filament-wound reinforced plastic propellant lines for high-pressure service. The program should also demonstrate the accuracy with which present techniques for the design and performance prediction of filament-wound structures may be extrapolated to cover the more complex high-pressure fluid transfer line.

The recommended approach would entail the use of available techniques for the design of low-pressure ducting and high-pressure toroidal pressure vessels and solid rocket motor cases to design subscale models of sample propellant lines. At least two different diameters (the smallest at least 4 in. dia) should be used to provide a verification of scaling factors. A relatively simple, single-plane configuration with at least two bends of different radius would provide a basis upon which several materials, such as "S" glass, "E" glass and Dacron could be compared, without the necessity of designing a more complex winding machine than that presently used for winding aircraft ducts. The machine used should provide sufficient flexibility to permit the use of various wrap angles in order that the effect of varying this parameter may also be demonstrated.

The testing of the sample lines should provide for proof testing (no permanent physical change in the sample) at 10,000 psi (6600-psi operating pressure with a 1.5 factor of safety) and demonstration of flexibility or bending characteristics at both 6600 psi and unpressurized. The unpressurized tests should be conducted both at ambient and LH_2 temperatures. The pressurization requirement will necessitate the use of liners and connectors. The liner material selection should be based upon the best information available at the time of fabrication, resulting from present studies. Connectors will have to be adaptable to removable test facility fixtures, but should provide criteria applicable to the attachment of flightweight connectors to the filament-wound structure. Effects of external fire should be evaluated.

VIII, H, Recommended Filament-Wound Line Feasibility Demonstration Program (cont.)

The results of this demonstration program will establish the flexibility which may actually be expected from filament-wound high-pressure lines. This information, combined with the results of the large diameter high-pressure bellows demonstration described above, will permit an accurate evaluation of the feasibility of developing high-pressure filament-wound lines for large rocket engines. In addition, the experience gained with lines and connectors will indicate the criticality of these problem areas.

IX. HEAT EXCHANGERS

The purpose of this study was to establish design criteria and to investigate the problems associated with heat exchangers that supply tank pressurant aboard advanced liquid rocket vehicles; emphasis was placed on those heat exchangers involving engine pump discharge pressures up to 6000 psia. A discussion of the propellant tank pressurization requirements is presented in Section VI,B,3,c. Design criteria were established by first analyzing the heat exchanger requirements during transient as well as steady-state operation. The significant factors affecting heat exchanger performance were then treated as variables in a parametric analysis. The results of the parametric analysis subsequently were used to design a heat exchanger for a specific application. The specific design analysis for this exchanger appears in Appendix H.

The results of this study indicate that no major technical problems exist in the design of pressurant heat exchangers for advanced liquid rocket vehicles having engine pump discharge pressures and thrust levels up to 6000 psia and 24×10^6 lb, respectively.

IX, Heat Exchangers (cont.)

NOMENCLATURE FOR HEAT EXCHANGER CALCULATIONS

Symbols	Definition
A	Cross-sectional area of shell, in.^{2}
A_c	Minimum free flow area across bank of tubes, in.^{2}
A_s	Inside area of heat transfer surface, in.^{2}
A_x	Cross-sectional area of tube, in.^{2}
c_p	Specific heat at constant pressure, Btu/lb-°R
D	Diameter of shell, in.
D_c	Average diameter of tubular coil, in.
d	Diameter of tube, in.
E	Modulus of elasticity, lb/in.^{2}
FS	Engine design safety factor, dimensionless; FS_w for combined engine design and tube bend safety factors
f	Friction factor, dimensionless; f_s for flow inside straight drawn-tubing; f_i for flow inside coiled tubing; f_o for flow normal to a bank of tubes
G	Mass velocity, lb/hr-ft^2 of cross section; for flow inside tubes, $G_i = w_l/A_x$; for flow across tubes, $G_o = w_g/A_c$
g_c	Conversion factor in Newton's law; g_c = 32.2 (lb fluid) (ft)/(sec^2) (lb force)
h	Local coefficient of heat transfer, Btu/hr-in.^{2}-°R
K	Universal gas constant, ft-lb/lb-mole-°R
k	Thermal conductivity, Btu-in./hr-ft^2-°R for tube material; Btu/hr-ft-°R for fluid data
L	Length of shell, in.
L_B	Length of tubing per bank ($L_B = L_T/N$), in.

IX, Heat Exchangers (cont.)

NOMENCLATURE FOR HEAT EXCHANGER CALCULATIONS (cont.)

Symbols	Definition
L_T	Total length of tubing, in.
LMTD	Logarithmic-mean-temperature difference, °R
mw	Molecular weight, gm/gm mole
N	Number of tubes, dimensionless
P	Pressure, psia
ΔP	Pressure loss, psia
Q	Rate of heat transfer, Btu/hr
q	Heat transfer rate per unit surface area, Btu/hr-in.^{2}
R	Local heat transfer resistance, hr-in.^{2}-°R/Btu
r	Radius of tube, in.
T	Absolute temperature, °R
ΔT	Absolute temperature differential, °R
U	Overall heat transfer coefficient, Btu/hr-in.^{2}-°R
V	Velocity of fluid, ft/sec
W	Weight, lb
w	Mass rate of flow, lb/sec
X_l	Ratio of longitudinal pitch to outside diameter of tubes, dimensionless
X_t	Ratio of transverse pitch to outside diameter of tubes, dimensionless

IX, Heat Exchangers (cont.)

NOMENCLATURE FOR HEAT EXCHANGER CALCULATIONS (cont.)

Symbols	Definition
GREEK LETTER SYMBOLS	
α	Thermal expansion coefficient of material, in./in.
γ	Ratio of specific heats, dimensionless
ε	Absolute roughness of tube wall surface, in.
μ	Dynamic viscosity of fluid, lb/ft-sec; μ_s for oxygen at an average inside tube wall temperature
ν	Poisson's ratio of material, dimensionless
π	3.14, dimensionless
ρ	Density, lb/ft^3
δ_y	Yield strength of material at 0.2 percent offset, $lb/in.^2$
τ	Tube wall thickness, in.
ϕ	Shell wall thickness, in.
DIMENSIONLESS VALUES	
M	Mach number
Pr	Prandtl number ($3600 \mu c_p/k$), a fluid property modulus
Re	Reynolds number ($\rho V d/12 \mu$), a flow modulus
SUBSCRIPTS	
b	Bulk
ex	Exit
f	Film
g	Hot gas

IX, Heat Exchangers (cont.)

NOMENCLATURE FOR HEAT EXCHANGER CALCULATIONS (cont.)

Symbols	Definition
SUBSCRIPTS (cont.)	
i	Inside
in	Inlet
l	Pressurant
o	Outside
w	Tube wall

IX, Heat Exchangers (cont.)

A. FACTORS INFLUENCING HEAT EXCHANGER REQUIREMENTS

An investigation of the heat exchanger requirements reveals that a number of factors have an important influence on the envelope, weight, fabricability, reliability, and cost of the heat exchanger. These factors are:

1. Fluid Parameters

a. Type of fluids
b. Chemical compositions
c. Flow rates
d. Temperature variations of each fluid
e. Pressure variations of each fluid
f. Transport properties
g. Flow configurations (e.g., counter flow, crossflow, etc.)
h. Differences between fluid temperatures
i. Differences between fluid pressures

2. Engineering Factors

a. Locations of unit(s)
b. Number of units
c. Surface area configurations
d. Flow arrangement (i.e., in the case of a tubular surface area configuration, which fluid flows inside the tubes)
e. Tube configurations
f. Tube arrangements, including spacings
g. Materials
h. Wall thickness criteria
i. Tube diameters
j. Envelope limitations
k. Number of sections and banks per section

IX, A, Factors Influencing Heat Exchanger Requirements (cont.)

l. Support structures

m. Weights

n. Fabrication techniques

o. Costs

3. Environmental Factors

a. Temperatures

b. Pressures

c. Materials

d. Acceleration

e. Vibration

f. Noise

A number of these factors do not vary once an engine system configuration has been selected. It is the variable factors that are at the discretion of the heat exchanger designer to manipulate until the optimum design is obtained.

The factors common to high-pressure engine systems are tabulated for steady-state operation in Table IX-A-1 for each of the six engine concepts outlined for investigation in this program. These engines were selected to cover the range of engine design parameters expected for future rocket propulsion systems. The primary purpose of the heat exchanger work is to determine whether technology problem areas exist in the design of heat exchangers for large-thrust high-pressure rocket engines. The parameter study effort covers a range of variables (fluid properties, tube sizes, tube thicknesses, fluid velocities, etc.) anticipated for the engine thrusts and pressures considered. In order to determine whether or not technology problem areas associated with detailed design and fabrication exist, a complete design study was performed for a representative engine. The AJ-1 engine was chosen for the specific design effort since it was the nominal engine for the total study effort.

TABLE IX-A-1

BASIC ENGINE CHARACTERISTICS PERTINENT TO HEAT EXCHANGER DESIGN

Engine Concept	Thrust Level lb x 10^6	Number of Chambers	Type of Plumbing System	Type of cycle	Thrust Chamber Pressure, psia	PROPELLANTS									PRIMARY COMPUSTOR/GGA COMBUSTED-GASES					
						FUEL				OXIDIZER					Total Flow Rate, lb/sec	Mixture Ratio	Chamber Pressure, psia	Turbine Inlet		Turbine Exit	
						Type	Pump Discharge		Flow Rate, lb/sec	Type	Pump Discharge		Total Flow Rate, lb/sec								
							Pressure, psia	Temp., °R			Pressure, psia	Temp., °R						Pressure, psia	Temp., °R	Pressure psia	Temp., °R
AJ-1	6	Multiple	Single	Staged Combus-tion	2500	LH_2	4550	37	2240	LO_2	4100	167	13430	4285	0.915	3620	3500	1900	2735	1800
AJ-2 (1st Version)	24	"	"	"	2500	LH_2	4680	37	8950	LO_2	4100	167	53710	17143	0.915	3650	3620	1900	2730	1800
AJ-2 (2nd Version)	24	"	"	"	2500	LH_2	4900	37	8950	LO_2	4800	167	53710	12828	1.050	4350	4270	1900	2760	1760
AJ-3	6	"	"	"	2500	RP-1	4000	541	5500	LO_2	4000	167	14300	14676	38.000	3670	3590	1770	2700	1685
AJ-5	6	Single	Single	GGA	3400	LH_2	4100	37	2420	LO_2	4100	167	13212	545	1.050	3650	3570	1900	174	1168
AJ-6	6	"	"	Staged Combus-tion	2500	LH_2	4550	37	2240	LO_2	4100	167	13430	4285	0.915	3620	3500	1900	2735	1800
AJ-8	1.5 (Modular)	"	"	Gas-gas Staged Combus-tion	2500	LH_2	4930	37	578	LO_2	4930	167	3465	760	0.900	4190	4190	1620	3000	1420

IX, Heat Exchangers (cont.)

B. HEAT EXCHANGER PARAMETRIC ANALYSIS

The parametric phase of this study was conducted to examine the inter-relationships existing between the many factors influencing heat exchanger performance. The analysis required the selection of nonvariable and variable parameters. The selected values and ranges of values for these two categories of parameters are presented and discussed in succeeding paragraphs. Culmination of this analysis is represented by the curves (Figures IX-B-1 through -16) relating the variable parameters to the overall heat transfer coefficient and weight of the tube bundle.

1. Nonvariable Parameter Selection

Where possible, these parameters were selected as those characteristic of the AJ-1 engine; for example, in the selection of a location for the exchanger unit only the AJ-1 engine configuration was considered. Other selected nonvariable parameters such as pressurant exit temperatures from the heat exchanger are independent of the engine operating characteristics and were based on characteristics of heat exchanger designs for present liquid rocket engines (Titan, F-1, J-2, M-1, etc.).

a. Location of Unit

The AJ-1 engine provides several possible locations for a heat exchanger unit. The most reasonable of these are (1) the secondary combustor coolant jacket, (2) the primary combustor to turbine (turbine inlet) duct, and (3) the turbine to secondary combustor (tubine exhaust) duct. The coolant jacket is a "ready-made" hydrogen heat exchanger. If the jacket were designed to heat the hydrogen to a sufficient temperature for tank pressurization, a small percentage of the total flow could be bled from the jacket and piped directly to the hydrogen tank. Since an engine heat-transfer analysis is beyond the scope of this program,

IX, B, Heat Exchanger Parametric Analysis (cont.)

only the turbine inlet or turbine exhaust ducts were considered. An analysis (Section VI, B, 3, e) showed that both of these locations are acceptable and provide similar conditions for heat exchanger operation. Therefore, it was not necessary to choose between the two for purposes of the parametric analysis.

b. Fluid Types

The fluid types considered in this analysis are representative of the various propellant tank pressurant fluids. These fluids can be divided into three broad categories: (1) propellant combustion products, (2) stored inert gases, and (3) vaporized propellants. The former method generally requires cooling the pressurant, while the latter two methods generally require heating the pressurant. The selection of the best method for a particular vehicle system is dependent primarily on the propellants used.

Heat exchanger requirements for the propellant combustion products method of tank pressurization were not considered because of the limited application of this technique with RP-1 and hydrogen fuels. Fuel tank contamination and coking of the heat exchangers and lines are problems with RP-1 gas generator products. The use of combustion products in cryogenic tanks involves the problem of pressurant condensation (H_2O) and consequent contamination of the propellants.

The stored inert gas method has found extensive application aboard vehicles using hydrocarbon propellants such as RP-1, because these propellants decompose on combustion and/or vaporization. Two stored gases that are most commonly used as pressurants are helium and nitrogen.

The vaporized propellant method has found extensive application for tank pressurization, especially for autogenous (self-generating) systems. This method, which is highly dependent on engine operating pressures, provides rigid design requirements for heat exchangers. Therefore, this method was selected for evaluation utilizing AJ-1 engine propellants (liquid hydrogen and liquid oxygen) as tank pressurants.

The selected location of the exchanger in the AJ-1 turbine inlet or exhaust ducts dictated the use of primary combustor exhaust gases (hereafter referred to as hot gas) as the source of heat for vaporizing the propellants. These exhaust gases are the combustion products of liquid hydrogen and liquid oxygen at a mixture ratio of 0.915.

c. Fluid Flow-Rates

Since detailed specific vehicle requirements were not established for this study, the pressurant flow rates selected represent the same percentage values used for the M-1 engine heat exchanger design: approximately two percent of the total liquid hydrogen flow rate (44 lb/sec) and one percent of the total liquid oxygen flow rate (132 lb/sec). By locating the exchanger in the turbine inlet or exhaust ducts, all the hot gas flow was available for use in the heat exchanger.

d. Fluid Temperature-Gradients

The heat exchanger inlet temperatures selected for the pressurants are the pump discharge temperatures for the AJ-1 engine. The selected exit pressurant temperatures from the heat exchanger are based on a review of present liquid rocket vehicles that utilize the same vaporized propellants (liquid hydrogen and liquid oxygen) for pressurization. This review, which encompassed such vehicles as Titan I, NOVA, and Saturn, indicated that 560°R and 960°R for the

IX, B, Heat Exchanger Parametric Analysis (cont.)

vaporized liquid hydrogen and liquid oxygen, respectively, were reasonable exit temperatures. To establish more accurate values for a given vehicle would require a better knowledge of both the vehicle configuration and propellant utilization schedule.

The hot gas temperature at the AJ-1 engine turbine exhaust (1800°F) was selected as the inlet temperature of the hot gas. The hot gas exit temperature was determined from a heat balance between the two fluids.

e. Heat Exchanger Surface Area Configurations

Two types of surface area configurations were considered for the heat exchanger: jacket and tubular. An examination of the jacket configuration as opposed to a tubular one revealed that it results in a larger envelope and requires a larger wall thickness. The thicker wall means added resistance to effective heat transfer and added heat exchanger weight. Therefore, major emphasis in this analysis was placed on the tubular configuration.

f. Tube Configuration

Three tube configurations were considered for this analysis: (1) flattened or dimpled tubes, (2) finned tubes, and (3) round tubes. The advantages and disadvantages of flattened or dimpled tubes and finned tubes relative to round tubes were considered. Flattened or dimpled tubes and finned tubes present a variety of potential shapes, the actual selection being dependent upon a trade-off between heat transfer advantages and the structural constraints imposed upon bending of the tube. Both configurations provide more effective heat transfer under certain conditions, but do not necessarily provide overall weight benefits. The flattened or dimpled tubes provide local increases in the pressurant velocity and resulting heat transfer benefits. However, these heat transfer benefits are better realized through use of tubes of smaller diameters where large pressurant

IX, B, Heat Exchanger Parametric Analysis (cont.)

pressure losses are allowable. The finned tubes, which provide heat transfer benefits through use of an extended surface area, increase costs and hot gas pressure losses. From the technology standpoint, flattened or dimpled tubes and finned tubes do not differ significantly from round tubes which lend themselves to a simpler analysis. Therefore, round tubes were selected for this analysis.

Pressure and heat exchanger packaging considerations resulted in selecting the pressurant to flow inside the tubes and the hot gas to flow perpendicularly across the tubes. This flow arrangement, commonly known as crossflow, provides minimum pressure losses per unit of heating surface area.* In addition, this arrangement eliminates most of the heater design problems associated with other flow arrangements such as counterflow, parallel flow, etc.**

g. Tube Material

The selection of tube material required consideration of various material properties. A list of possible tube materials, all of which are compatible with the fluids under consideration, and their physical properties is presented in Table IX-B-1.

The materials were compared to determine those that would provide maximum heat transfer per unit weight. Material selection made on this basis provides the lightest, most compact tube bundle. The heat transfer capability of a tube is directly proportional to the thermal conductivity per unit wall thickness of the tube, i.e.,

$$q \sim \frac{k}{\tau}$$

* E. R. G. Eckert and R. M. Drake, Jr., Heat and Mass Transfer, McGraw-Hill Book Company, Inc., New York, 1959.

** R. A. Stevens, Mean Temperature Difference in Multi-pass Crossflow Heat Exchangers, Convair Report AD 400838, September 1955 (Unclassified).

IX, B, Heat Exchanger Parametric Analysis (cont.)

The tube weight is directly proportional to its density and wall thickness, i.e.,

$$W \sim \rho \tau$$

Thus maximum heat transfer per unit weight is directly proportional to $k/\rho \tau^2$ where k and ρ are material properties and τ is a function of the loading on the tube wall. From this relationship it is readily apparent that the minimum wall thickness is desired. Designing for minimum wall thickness requires consideration of both tensile and compressive loading.

The wall thickness required for tensile loading is represented by

$$\tau = \left[\frac{FS_w \, d_i}{2 \, \sigma_y} \right] (P_i - P_o) \qquad (3)^*$$

and for compressive loading by

$$\tau = \frac{d_i}{2} \left[\sqrt[3]{\frac{FS_w (1-\nu^2) \, (P_o - P_i)}{0.25 \quad E}} \right] \qquad (4)^*$$

For tensile loading (Equation 3), the wall thickness is indirectly proportional to the material's yield strength. In compressive loading (Equation 4), the wall thickness is directly proportional to the 2/3 power of the material's Poission ratio and indirectly proportional to the 1/3 power of the modulus of elasticity. The Poisson ratio was eliminated because it is constant at a value of 0.3 for the materials considered. Thus, in terms of material properties, the maximum heat transfer per unit weight is directly proportional to $\sigma^2_y \, k/\rho$ for tensile

* R. J. Roark, Formulas for Stress and Strain, McGraw-Hill Book Company, Inc. New York, 1964.

IX, B, Heat Exchanger Parametric Analysis (cont.)

loading and $E^{2/3} k/\rho$ for compressive loading. The material comparisions made on these relationships (designated tensile load index and compressive load index, respectively) are shown in Table VII-C-2.

Another basis for comparison was that of thermal stress index (TSI). A high TSI value represents a material's capability to withstand thermal stresses resulting from nonuniform temperature distribution. Figure IX-B-17 is a plot of TSI vs temperature. The thermal stress index is determined by

$$TSI = \frac{\nabla y \; k}{12 \; \alpha \; E} \tag{5}$$

Results of these comparisons (Table IX-B-1) indicate that Inconel 718 and René 41 are the best of the contending materials. The decision to use Inconel 718 for this study was based on: (1) its slightly higher TSI value and (2) the greater availability of data concerning its use as a tube material. René 41 has been used more widely for forgings and investment castings than as a tube material.

h. Additional Basic Considerations and Assumptions

This analysis also required the following additional considerations and assumptions:

(1) Tube diameters were constant between tube ends.

(2) Heat losses through the heat exchanger shell walls were negligible.

IX, B, Heat Exchanger Parametric Analysis (cont.)

(3) Outside tube wall temperatures were maintained above 500°R to avoid any possible condensation of water vapor from the hot gas.

(4) Scale deposits (fouling factors) were negligible.

(5) Bulk conditions of the fluids were assumed to be equal to the arithmetic mean of their limits.

(6) Overall heat transfer coefficients were calculated on the basis of the inside surface of the tubes.

2. Variable Parameters

The variable parameters in this analysis include those most influenced by engine operating pressures, i.e., pressurant and hot gas pressures; and three additional parameters, the pressurant and hot gas velocities and the tube diameter, which generally do not have fixed design values for a particular application. Any combination of pressurant velocity and tube diameter provides some measure of the required heat exchanger envelope once surface area requirements have been determined. A wide range of values for each parameter were considered including values characteristic of present liquid engines as well as advanced high-pressure engines such as the AJ-1 engine.

a. Pressurant Pressures (2000 to 6000 psia)

The range selected here encompasses pump discharge pressures of present engines such as the M-1 (approximately 2000 psia) and the advanced high-pressure engine concepts investigated in this program (3000 to 6000 psia).

IX, B, Heat Exchanger Parametric Analysis (cont.)

b. Hot Gas Pressures (1500 to 5500 psia)

Review of the basic engine characteristics (Table IX-A-1) indicated that the differential pressure between hot gas pressures and their corresponding pump discharge pressures is approximately 500 psia.

c. Pressurant Velocities (0 to 1500 ft/sec for hydrogen; 0 to 450 ft/sec for oxygen)

Compressible gas flow studies * indicate that a maximum velocity corresponding to Mach 0.3 provides a margin of safety for unknown flow discontinuities which could cause the velocity to become sonic and thereby choke the flow. Therefore, the maximum allowable pressurant velocity is represented by

$$V_1 = 0.3\sqrt{\frac{g_c \gamma K T}{mw}}$$

The fluid properties remain relatively constant over the temperature and pressure ranges under consideration. Therefore, the maximum allowable velocity occurs at the highest temperature, i.e., the pressurant exit temperature. This value for each of the pressurants was selected as a nonvariable parameter (Section IX, B, 1) and, therefore, provides maximum allowable exit velocities (Mach 0.3) of 1320 ft/sec and 434 ft/sec for the hydrogen and oxygen, respectively.

* A. H. Shapiro, The Dynamics and Thermodynamics of Compressible Fluid Flow, Vol. I, The Ronald Press Company, New York, 1953.

G. K. Platt and C. C. Wood, Saturn Booster Liquid Oxygen Heat Exchanger Design and Development, Rocketdyne Report, August 1961 (Unclassified).

IX, B, Heat Exchanger Parametric Analysis (cont.)

d. Hot Gas Velocities (0 to 1680 ft/sec)

The maximum allowable hot gas velocity was also assumed to correspond to Mach 0.3. This limitation provided a maximum velocity of 1680 ft/sec at the highest temperature of the hot gas (1800°R).

e. Tube Inside Diameters (0.50 in., 0.75 in., and 1.00 in.)

In general, the size and weight of the tube bundle decreases with decreasing tube diameter until minimum gage wall thickness results. However, practical considerations of exchanger fabrication and complexity tend to favor larger tubes. The three tube diameters considered were selected after reviewing designs of present rocket engine heat exchangers for Titan, F-1, J-2, M-1, etc.

3. Calculation Procedure

The overall heat transfer coefficients and tube bundle weights were calculated from the following equations:

a. Heat Balance

$$Q_1 = 3.6 \times 10^3 \, w_1 \, c_{p_1} \, (T_{ex} - T_{in})_1 \qquad (7)$$

$$= 3.6 \times 10^3 \, w_g \, c_{p_g} \, (T_{ex} - T_{in})_g \qquad (8)$$

b. Hot Gas Exit Temperature

$$T_{ex_g} = T_{in_g} - \frac{Q_1}{3.6 \times 10^3 \, w_g \, c_{p_g}} \qquad (9)$$

IX, B, Heat Exchanger Parametric Analysis (cont.)

c. Logarithmic-Mean-Temperature Difference (crossflow configuration)

$$LMTD = \frac{\left(T_{ex_g} - T_{in_l}\right) - \left(T_{in_g} - T_{ex_l}\right)}{\ln\left[\dfrac{T_{ex_g} - T_{in_l}}{T_{in_g} - T_{ex_l}}\right]} \qquad (10)$$

The logarithmic-mean-temperature difference was applied because the temperature of the hot gas remains essentially constant across the exchanger.*

d. Average Coefficients of Heat Transfer

$$h_l = 0.023 \frac{k_l}{12\, d_i} \left[Re\right]_b^{0.8} \left[Pr_l\right]_b^{0.4} \left[\frac{T}{T_w}\right]_b^{0.34} \quad \text{Hydrogen} \qquad (11)^{**}$$

$$h_l = 0.027 \frac{k_l}{12\, d_i} \left[Re\right]_b^{0.8} \left[Pr_l\right]_b^{1/3} \left[\frac{\mu}{\mu_s}\right]_b^{0.14} \quad \text{Oxygen} \qquad (12)^{*}$$

$$h_g = 0.0348 \frac{k_g}{12\, d_o} \left[Re\right]_f^{0.805} \left[Pr_g\right]_f^{1/3} \quad \text{Hot Gas} \qquad (13)^{***}$$

The average heat transfer coefficients for the pressurants (Equations 11 and 12) are based on bulk temperatures while the coefficient for the hot gas (Equation 13) is based on the film temperature. To determine the film

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* W. H. McAdams, Heat Transmission, Third Edition, McGraw-Hill Book Company, Inc., New York, 1954.

** R. W. Thompson and E. L. Geery, Heat Transfer to Cryogenic Hydrogen at Supercritical Pressures, Aerojet-General Report 1842, July 1960 (Unclassified).

*** B. Gebhart, Heat Transfer, McGraw-Hill Book Company, Inc., New York, 1961.

IX, B, Heat Exchanger Parametric Analysis (cont.)

temperature it was necessary to first determine inside and outside tube wall temperatures. The results of sample calculations indicated that inside and outside tube wall temperatures of 600 and 1600°R, respectively, were representative for this study. Furthermore the calculations indicated that these temperatures can deviate 200 to 300°R without significantly affecting the heat transfer coefficients.

Therefore, the preceding calculated temperatures (bulk for the pressurants and film for the hot gas) were used to determine the property data for the fluids (Table IX-B-2). The data for the pressurants were obtained from recently published Aerojet-General reports* on the two fluids; the data for the hot gas were obtained from a computer program that simulates combustion for shifting equilibrium considerations. The program showed that the hot gas (LO_2/LH_2 combustion products for a mixture ratio of 0.915) has the following combustion by volume:

Gaseous hydrogen	88.46%
Water	11.53%
Hydrogen ion (H+)	very small

*Properties of Para-Hydrogen, Aerojet-General Report 9050-65, June 1963 (Unclassified).

Properties of Oxygen, Aerojet-General Report 9200-11-63, 27 September 1963 (Unclassified).

IX, B, Heat Exchanger Parametric Analysis (cont.)

e. Overall Heat Transfer Coefficient (based on inside surface of tubes)

$$U = \frac{1}{R_1 + R_w + R_g} \tag{14}$$

where, $R_1 = 1/h_1$

$$R_w = \frac{144\ d_i \ \ln\ (d_o/d_i)}{2 \quad k_w}$$

$$R_g = (d_i/d_o)\ \frac{1}{h_g}$$

f. Inside Tube Surface Area

$$A_s = \frac{Q_1}{LMTD\ U} \tag{15}$$

g. Length of Tubing

$$L_T = \frac{A_s}{\pi\ d_i} \tag{16}$$

h. Tube Bundle Weight

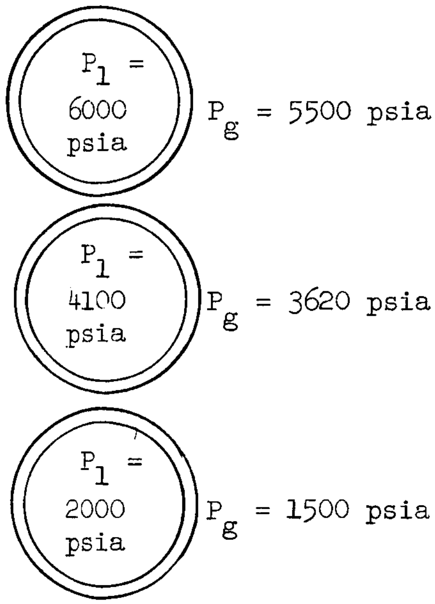
$$W = \frac{\pi \rho_w\ L_T}{4}\ (d_o^2 - d_i^2) \tag{17}$$

4. Discussion of Results

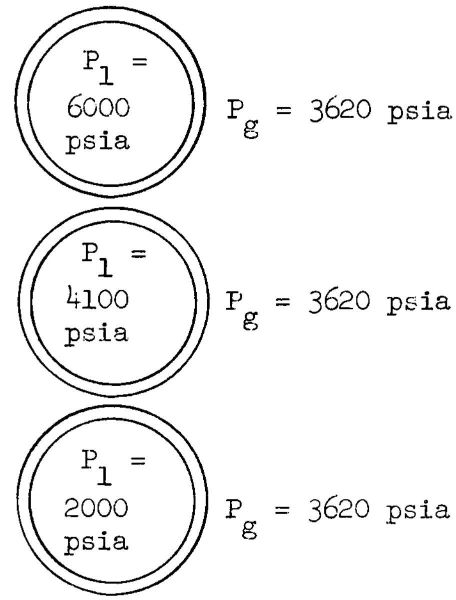
The results of this analysis are presented in Figures IX-B-1 through -16. The curves show the effects of variations in pressurant and hot gas pressures and velocities and in tube diameter on the overall heat transfer coefficient and tube bundle weight. These combined effects are shown for the following two-system pressure conditions:

IX, B, Heat Exchanger Parametric Analysis (cont.)

(1) Constant pressure differential of 500 psia across tube wall (Figures IX-B-1 through -6)



and (2) Constant hot gas pressure of 3620 psia (Figures IX-B-7 through -16)



(1) Constant Pressure Differential of 500 psia across Tube Wall

Combining Equations 11, 12, and 13 with 14 reveals that the overall heat transfer coefficient is a function of fluid velocities, fluid properties (ρ , μ, c_p, and k), tube wall property (k_w), and tube geometry (d_i and τ). The wall thickness (τ) is determined from Equations 3 and 4 for tensile and compressive loading, respectively. The higher value of τ represents the minimum allowable for use in the heat transfer equations. A wall thickness greater than the allowable minimum is undesirable because of the increased tube weight per unit length and increased heat transfer surface area. The increased surface area results in a larger tube bundle, which in turn, demands a corresponding enlargement of the shell and support structure with an accompanying increase in overall heat exchanger weight and volume.

IX, B, Heat Exchanger Parametric Analysis (cont.)

A comprehensive analysis of the AJ-1 engine pressure schedule during transient as well as steady-state operation (Table IX-B-3) was conducted to determine the pressure values for use in calculating the tube wall thickness. The analysis showed the nominal pressurant pressure, assuming a pressurant pressure equal to the pump discharge pressure, represents a maximum possible differential pressure. Therefore, tensile loading represents the more severe loading condition, and Equation 3 was used to determine the tube wall thickness.

Fluid properties can be determined if fluid velocities, temperatures, and pressures are specified. All temperatures are selected except for the hot gas exit temperature (T_{ex_g}) which is determined from the heat balance (Equation 9). The thermal conductivity of the tube wall is specified once the material (Inconel 718) has been selected. Therefore, the overall heat transfer coefficient can be evaluated as a function of V_l, V_g, P_l, P_g, and d_i (Figures IX-B-1 through -4). These figures indicate that increased heat transfer is obtained by increasing the fluid velocities. However, only small heat transfer benefits are obtained at the high velocities, and these benefits are partially offset by the rapidly increasing fluid pressure losses. Since added pressure loss for the turbine exhaust gases results in reduced engine performance, the efficiency relationship of the velocity versus heat transfer is significant.

These same curves also show that liquid hydrogen provides higher heat-transfer rates than liquid oxygen, which can be attributed primarily to the higher thermal conductivity and higher specific heat of the liquid hydrogen.

Figures IX-B-1 through -4 also indicate the effect of system pressures (defined here as engine systems having hot gas pressures equal to pressurant pressures minus 500 psia) on the overall heat transfer coefficient. This relationship shows that heat exchanger effects caused by system pressures

IX, B, Heat Exchanger Parametric Analysis (cont.)

become insignificant at pressure levels above P_1 = 2000 psia for hydrogen and P_1 = 4000 psia for oxygen.

By utilizing the results of Equations 7 and 10 together with the overall heat transfer coefficient (Figures IX-B-1 through -4) in equations 15, 16, and 17, the tube bundle weight of both the hydrogen and oxygen heat exchangers can be determined (Figures IX-B-5 and -6). These curves indicate that low system pressures and small tube diameters result in the lowest weight. Even at high system pressures, the tube bundle weight is a small fraction (approximately 0.3%) of the total engine weight.

(2) Constant Hot Gas Pressure of 3620 psia

The condition of constant hot gas pressure shows the effect of decreasing the pressurant pressure prior to entering the heat exchanger. This evaluation was considered valuable because the pressurant pressure is a critical factor in calculating tube wall thickness (Table IX-B-3). In the preceding case of constant pressure differential across the tube wall, the tube wall thickness was determined from tensile loading considerations (Equation 3). However, for a constant gas pressure of 3620 psia outside the tube, compressive loading considerations represent the more severe loading condition at pressurant pressures less than 3200 psia. Therefore, for these pressure conditions Equation 4 was used to determine the tube wall thickness. This is evident in Figures IX-B-7 and -8 which represent the local heat transfer resistance as a function of pressurant pressure, fluid velocity, and tube diameter for both hydrogen and oxygen heat exchangers. The wall resistance is a direct function of the wall thickness and, therefore, reflects the tube wall sizing considerations mentioned previously.

Combining the local resistances in Equation 14 results in Figures IX-B-9 and -10 for overall heat transfer coefficient. Again, the effect of tube wall sizing considerations is prominent. The optimum heat exchanger should operate as near the maximum U as possible.

IX, B, Heat Exchanger Parametric Analysis (cont.)

The variations of overall heat transfer coefficient with pressurant velocity, pressurant pressure, and tube size are shown in Figures IX-B-11 through -14. The curve trends are quite similar to those discussed for the constant pressure differential case.

Finally, Equations 15, 16, and 17 can be used to determine tube bundle weight (Figures IX-B-15 and -16). The effect of compressive consideration in sizing tube wall thickness has resulted in a definite minimum for tube bundle weight. Heat exchangers designed with constant hot gas pressure considerations should operate as near the minimum weight point as possible. This minimum point is also verified by the overall heat transfer coefficient (Figures IX-B-9 and -10). Again the tube bundle weight is a small percentage (approximately 0.15%) of the total engine weight.

All the figures reflect data indicating that the 0.5 in. diameter tube is the best of the three diameters when considered from the standpoint of maximum heat transfer per unit weight. The relative importance of this parameter becomes increasingly greater with higher pressures.

TABLE IX-B-1

COMPARISON OF TUBE MATERIALS

Tube Material	Density (ρ) lb/ft^3	*Thermal Conductivity (k) Btu-in./hr-ft^2-°R	Yield Strength at 0.2% Offset (σ_y) lb/in.^{2}	Modulus of Elasticity (E) lb/in.^{2}	Thermal Expansion Coefficient (α) in./in.	Tensile Load Index $\frac{\sigma_y^2 k}{\rho}$	Compressure Load Index $\frac{E^{2/3} k}{\rho}$	Thermal Stress Index $\frac{\sigma_y k}{12 \alpha E}$ Btu/hr-ft-°R
Inconel 718	0.290	110	125,000	23.5×10^6	8.4×10^{-6}	5.94×10^{12}	3.12×10^7	5.81×10^3
Stainless Steel 347	0.290	140	25,000	19.6×10^6	11.1×10^{-6}	0.30×10^{12}	3.50×10^7	1.34×10^3
René 41	0.298	109	120,000	25.0×10^6	8.21×10^{-6}	5.27×10^{-12}	3.13×10^7	5.32×10^3
Waspaloy	0.296	115	105,000	24.0×10^6	8.8×10^{-6}	4.29×10^{12}	3.23×10^7	4.77×10^3

* The thermal conductivity values are shown for an assumed average tube wall temperature of 1160°R. All other property values are shown for an assumed maximum outside tube wall temperature of 1860°R.

TABLE IX-B-2

FLUID PROPERTY DATA

Properties	Pressurant - LH_2 T_b = 300°R			Pressurant - LO_2 T_b = 565°R			Hot Gas - LO_2/LH_2 Combustion Products T_f = 1700°R		
	P_1 = 2000 psia	P_1 = 4100 psia	P_1 = 6000 psia	P_1 = 2000 psia	P_1 = 4100 psia	P_1 = 6000 psia	P_g = 1500 psia	P_g = 3620 psia	P_g = 5500 psia
mw gm/gm-mole	2.016	2.016	2.016	32.0	32.0	32.0	3.86	3.86	3.86
c_p Btu/lb-°R	4.12	4.20	4.25	0.261	0.355	0.375	1.9385	1.9385	1.9385
γ	1.40	1.40	1.40	1.40	1.40	1.40	1.354	1.354	1.354
k Btu/ft-hr-°R	0.093	0.102	0.107	0.0209	0.0275	0.0310	0.23928	0.23928	0.23928
μ lb/ft-sec	0.445×10^{-5}	0.505×10^{-5}	0.548×10^{-5}	1.675×10^{-5}	2.05×10^{-5}	2.43×10^{-5}	1.837×10^{-5}	1.837×10^{-5}	1.837×10^{-5}
ρ lb/ft³	1.12	2.10	2.77	10.93	22.48	29.66	0.3176	0.7664	1.1644
Pr	0.7097	0.7486	0.7836	0.7530	0.9530	1.058	0.5363	0.5363	0.5363
μ_s lb/ft-sec				1.82×10^{-5}	2.10×10^{-5}	2.64×10^{-5}			

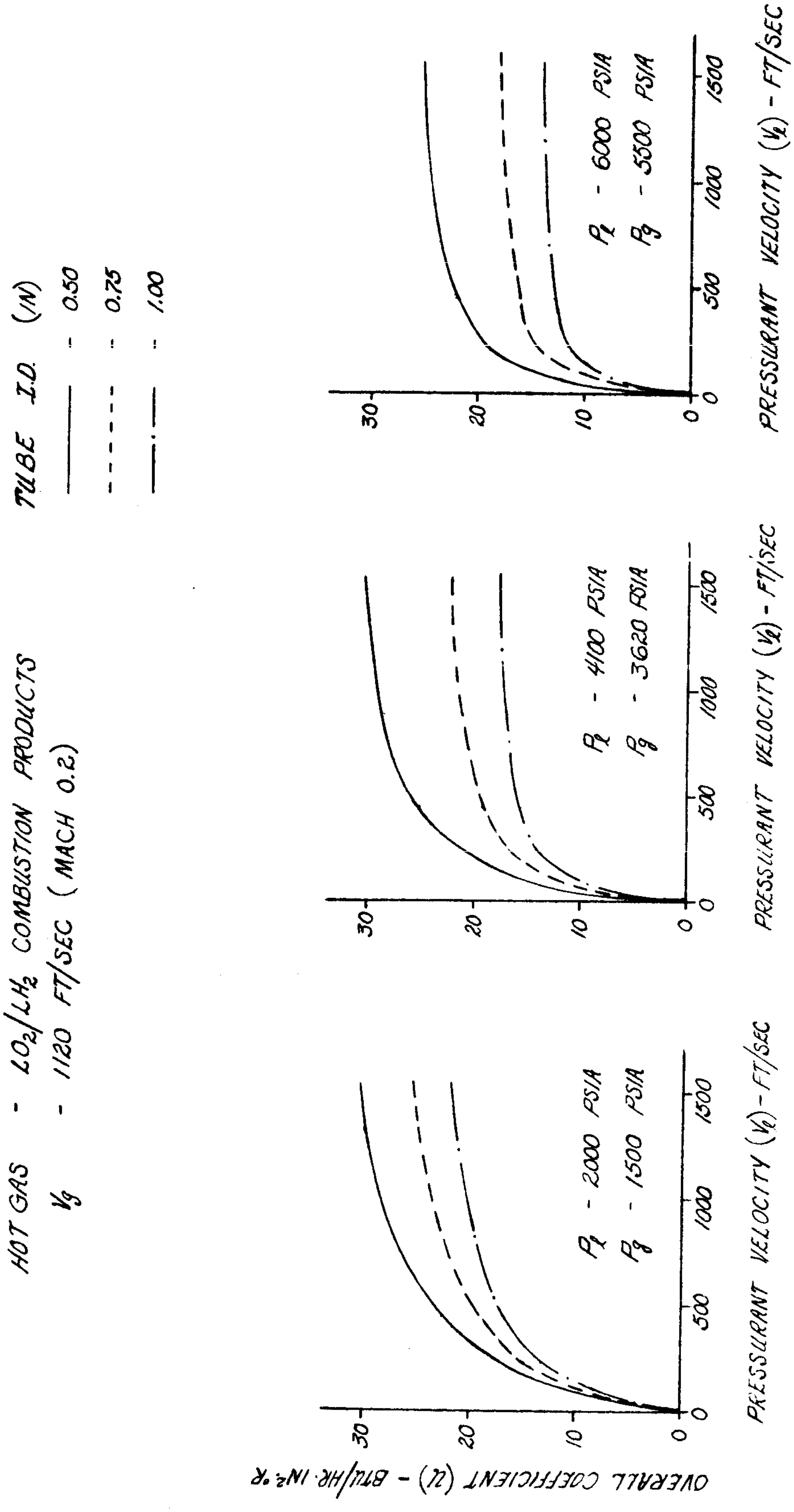
*NOTE: Evaluated at an assumed inside tube wall surface temperature of 700°R.

TABLE IX-B-3

HEAT EXCHANGER INLET PRESSURE SCHEDULE FOR AJ-1 ENGINE

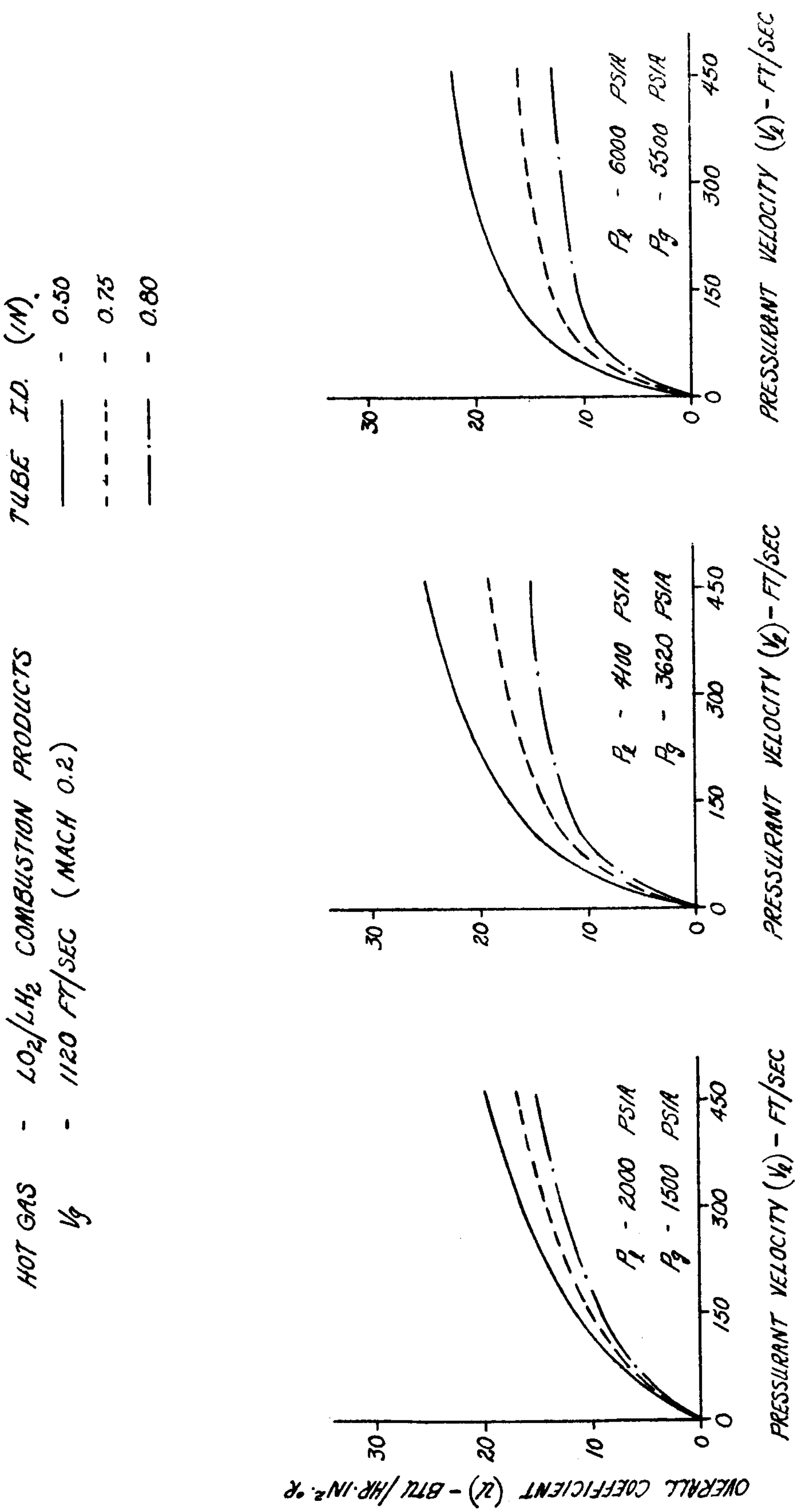
Pressure \ System Location / Pressurant		Turbine Inlet LH_2	Turbine Inlet LO_2	Turbine Exhaust LH_2	Turbine Exhaust LO_2
1. Nominal pressurant pressure during steady-state, psia	(P_1)	4500	4100	4500	4100
2. Maximum pressurant pressure during start transient, psia	(P_1)	5700	5450	5700	5450
3. Nominal hot gas pressure, psia	(P_g)	3500		2750	
4. Maximum hot gas pressure, psia	(P_g)	3600		2750	
5. Hot gas pressure at maximum pressurant pressure, psia	(P_g)	2700		1250	
6. Nominal differential pressure, psi $(P_{1-3} = P_1 - P_g)$		1000	600	1750	1350
7. Maximum differential pressure, psi $(P_{2-5} = P_1 - P_g)$		3000	2750	4450	4200

NOTE: All pressures are shown to the nearest 50 psi.



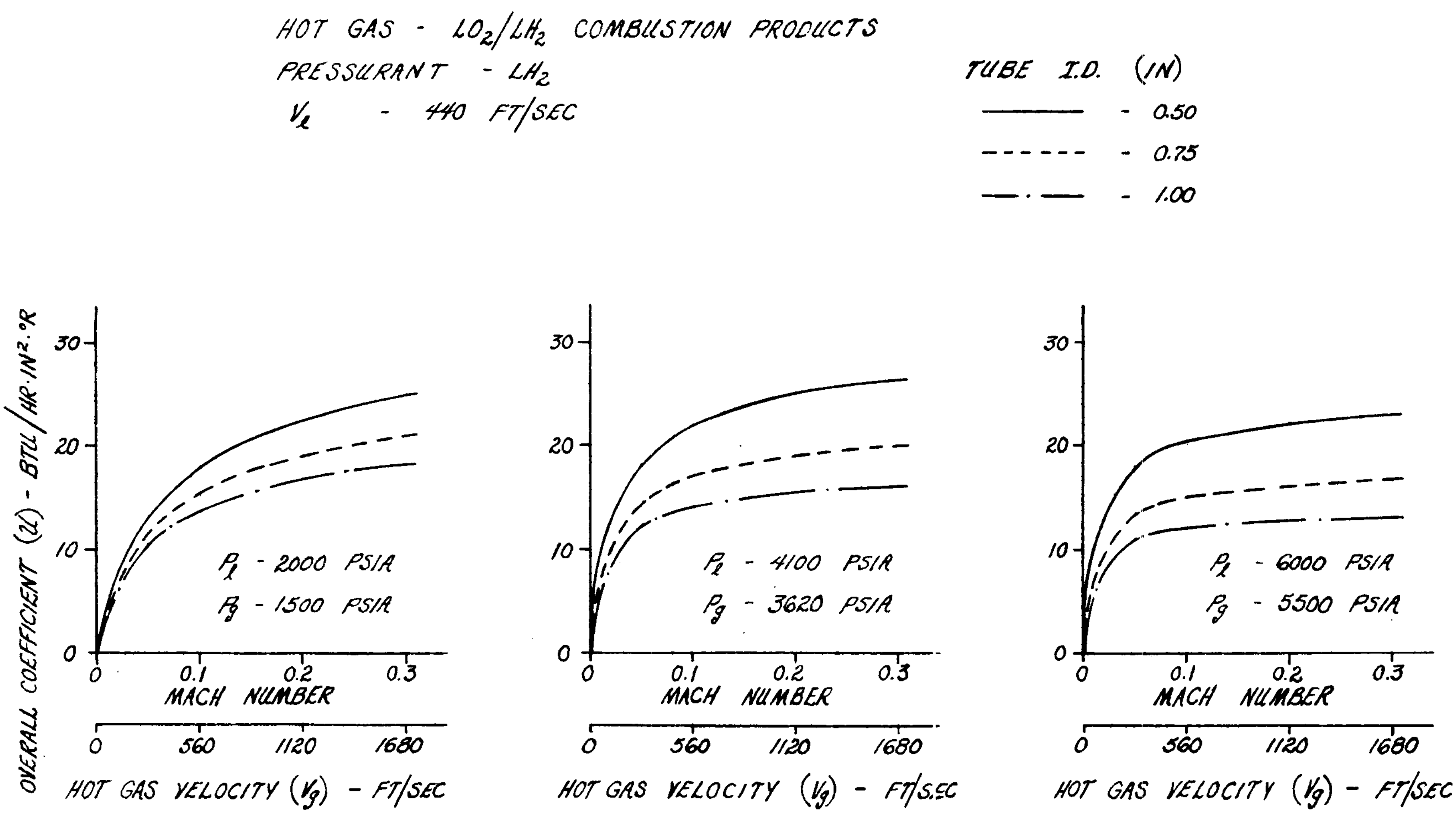
Overall Heat Transfer Coefficient vs Pressurant (LH_2) Velocity

Figure IX-B-1



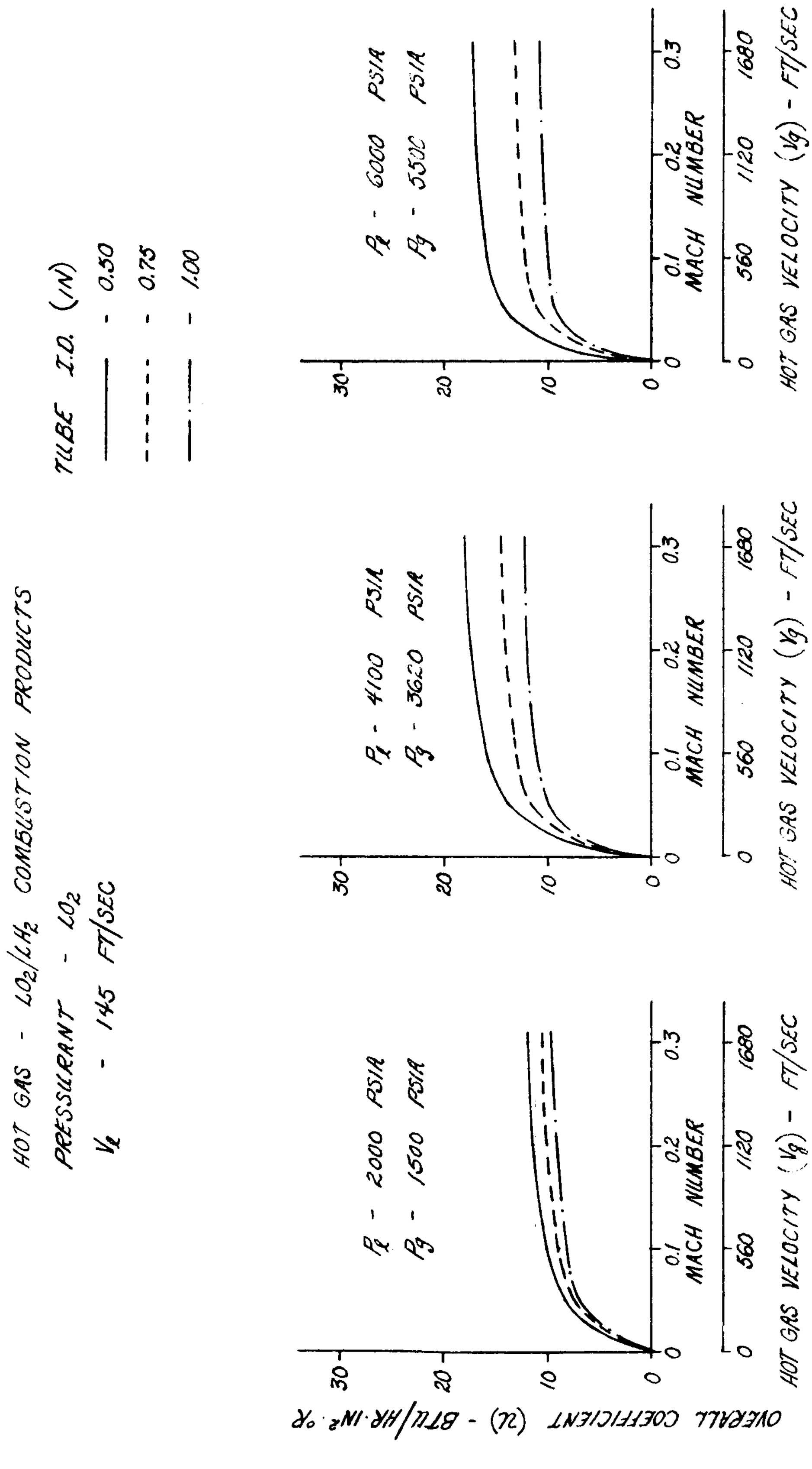
Overall Heat Transfer Coefficient vs Pressurant (LO_2) Velocity

Figure IX-B-2



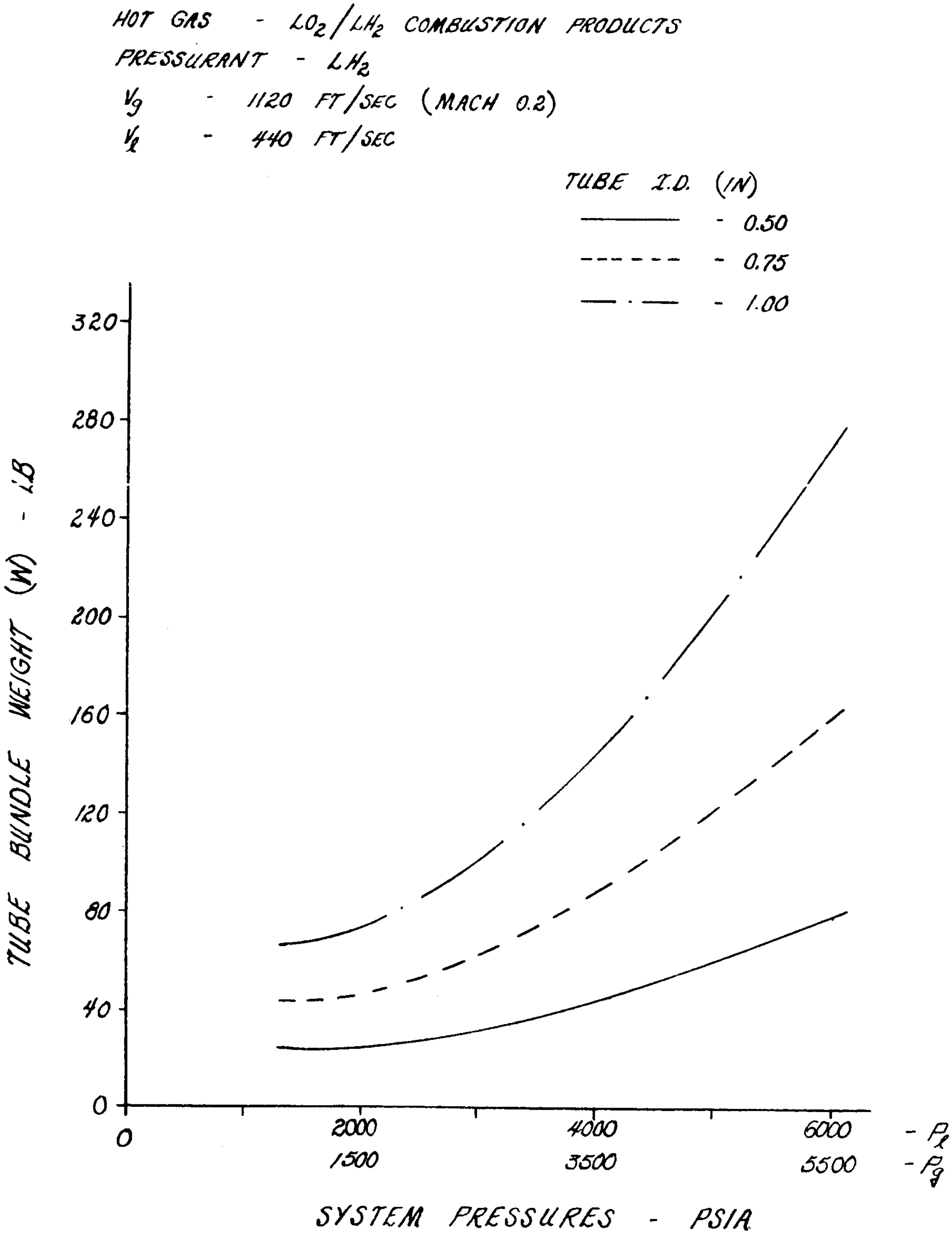
Overall Heat Transfer Coefficient vs Hot Gas Velocity

Figure IX-B-3



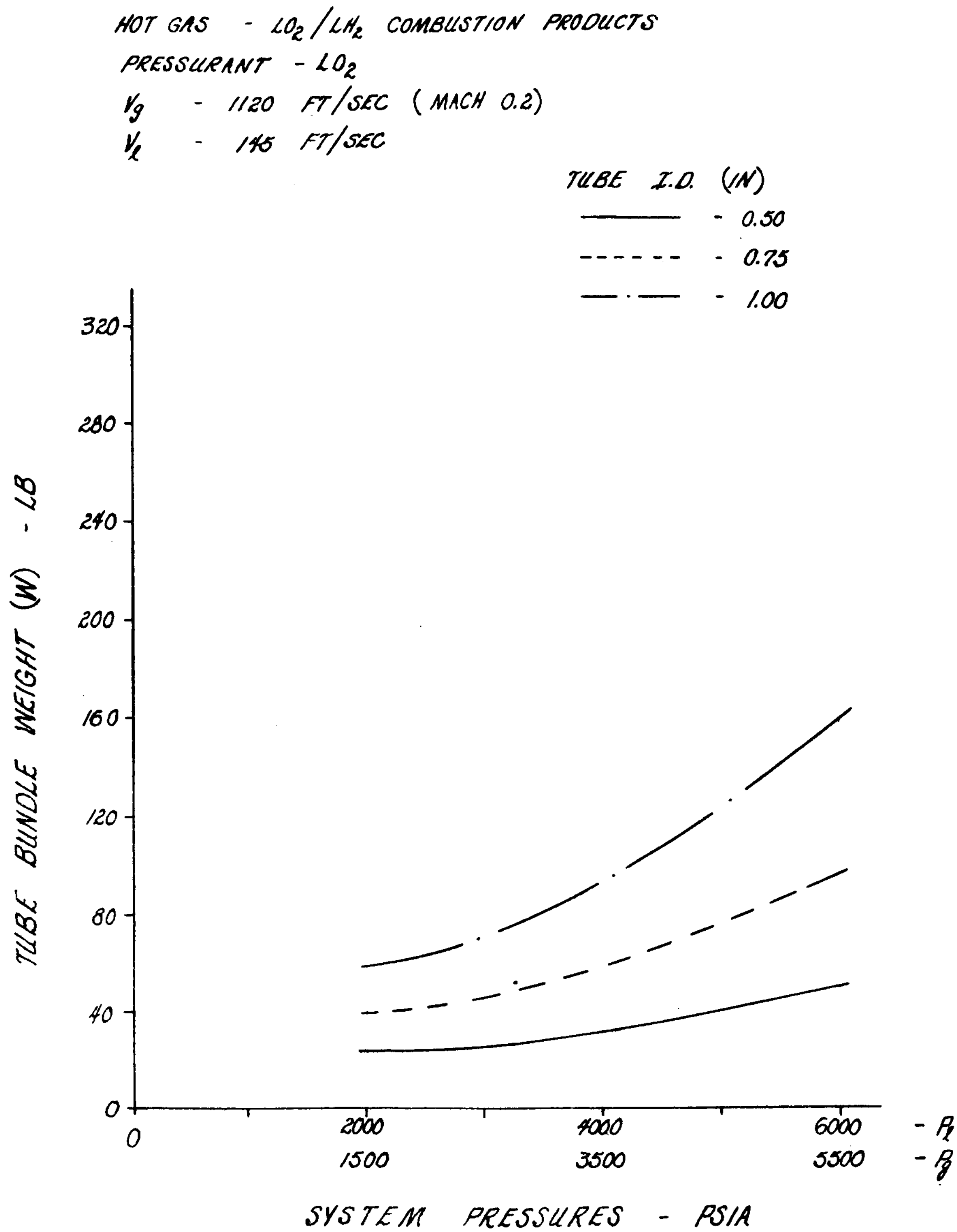
Overall Heat Transfer Coefficient vs Hot Gas Velocity

Figure IX-B-4



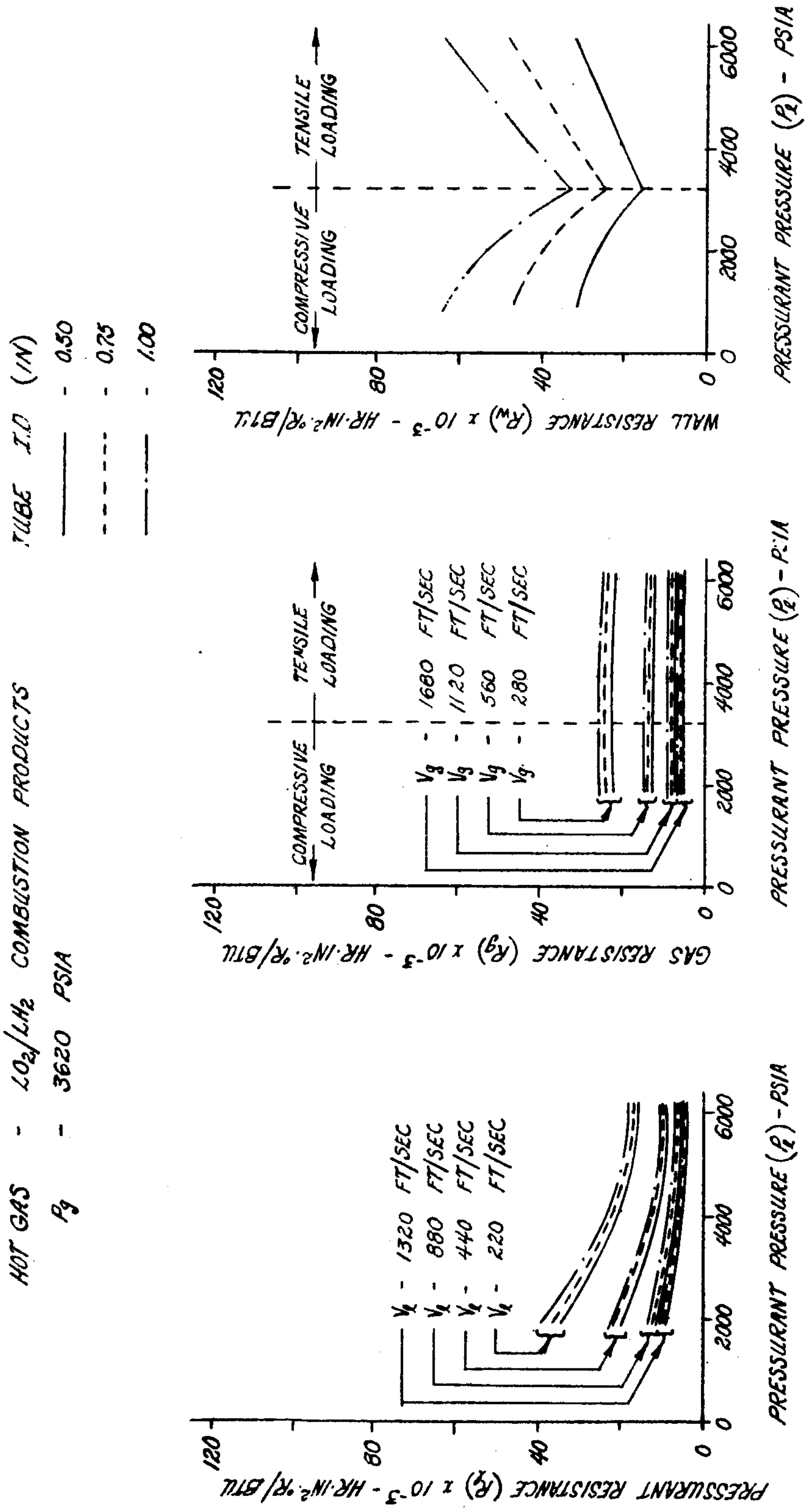
Tube Bundle Weight vs System Pressures

Figure IX-B-5



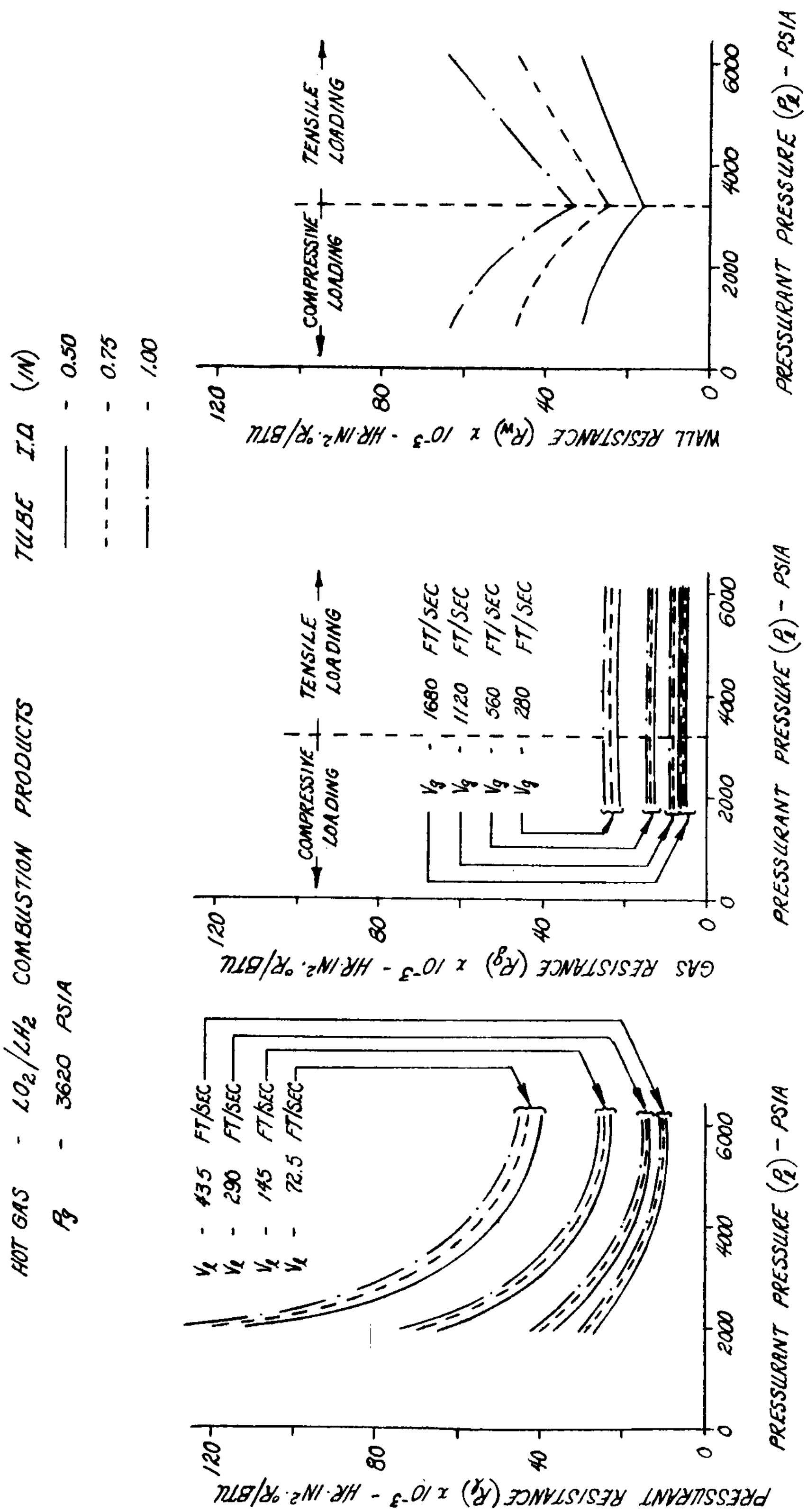
Tube Bundle Weight vs System Pressures

Figure IX-B-6



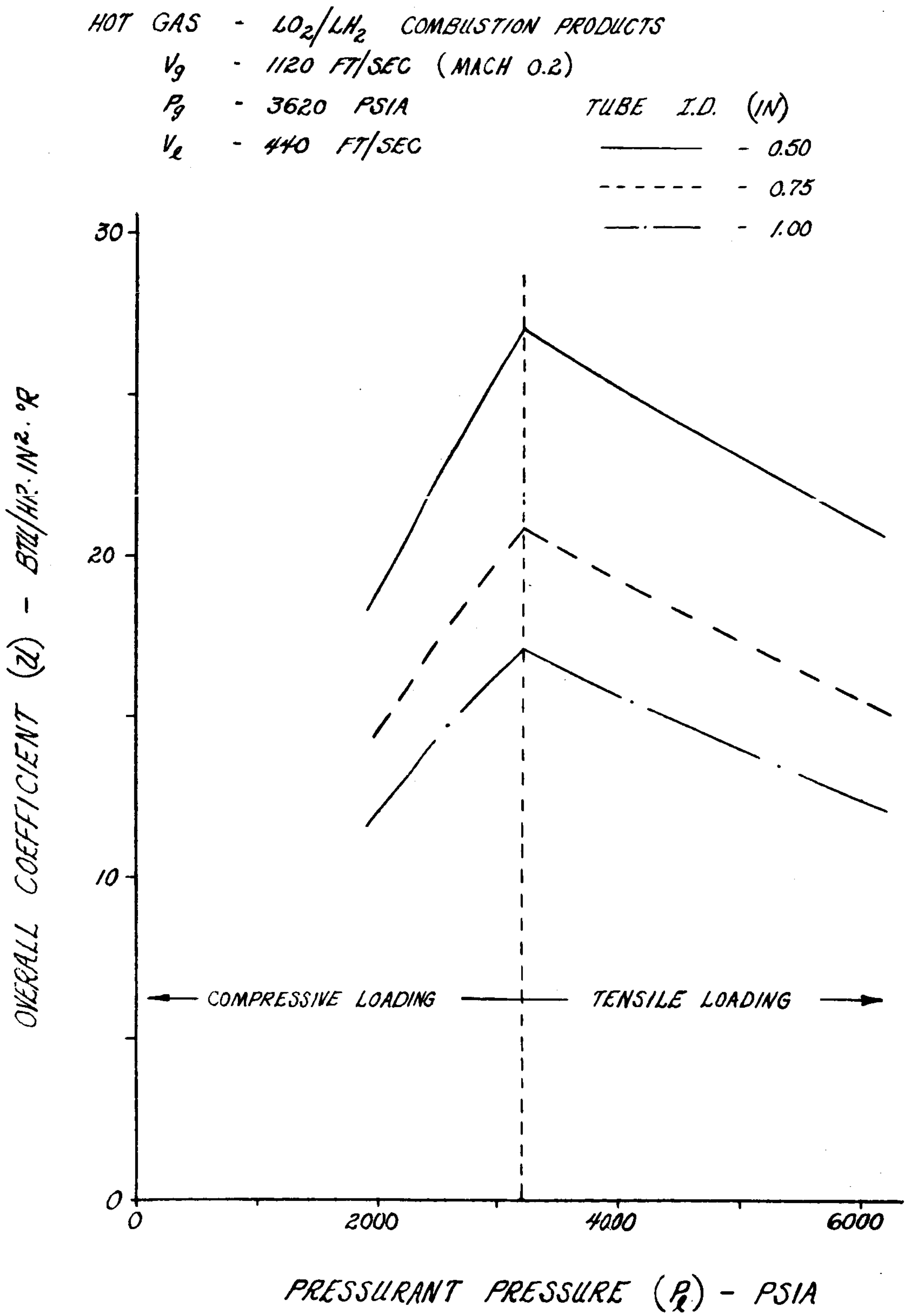
Local Heat Transfer Resistances vs Pressurant (LH_2) Pressure

Figure IX-B-7



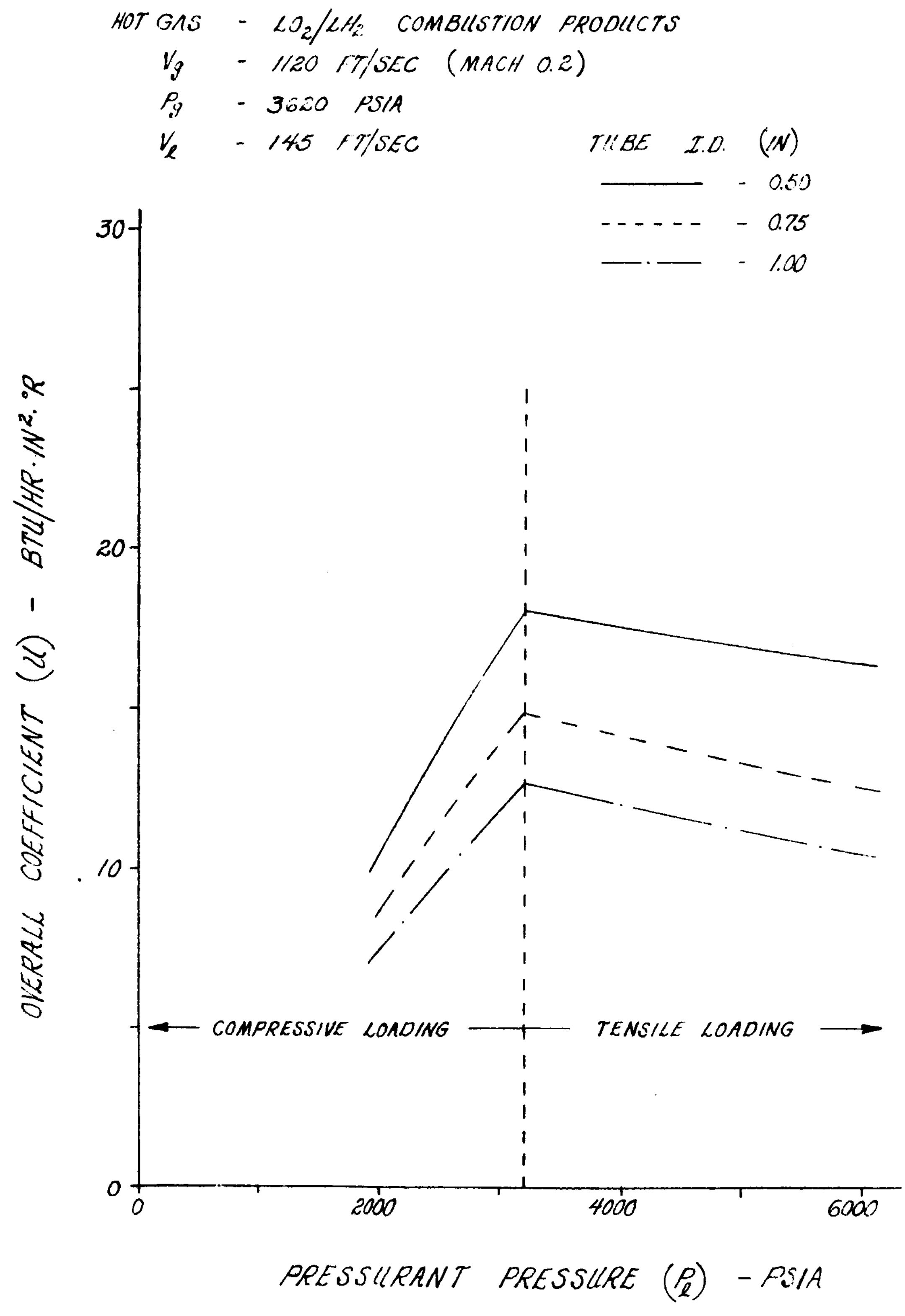
Local Heat Transfer Resistances vs Pressurant (LO_2) Pressure

Figure IX-B-8



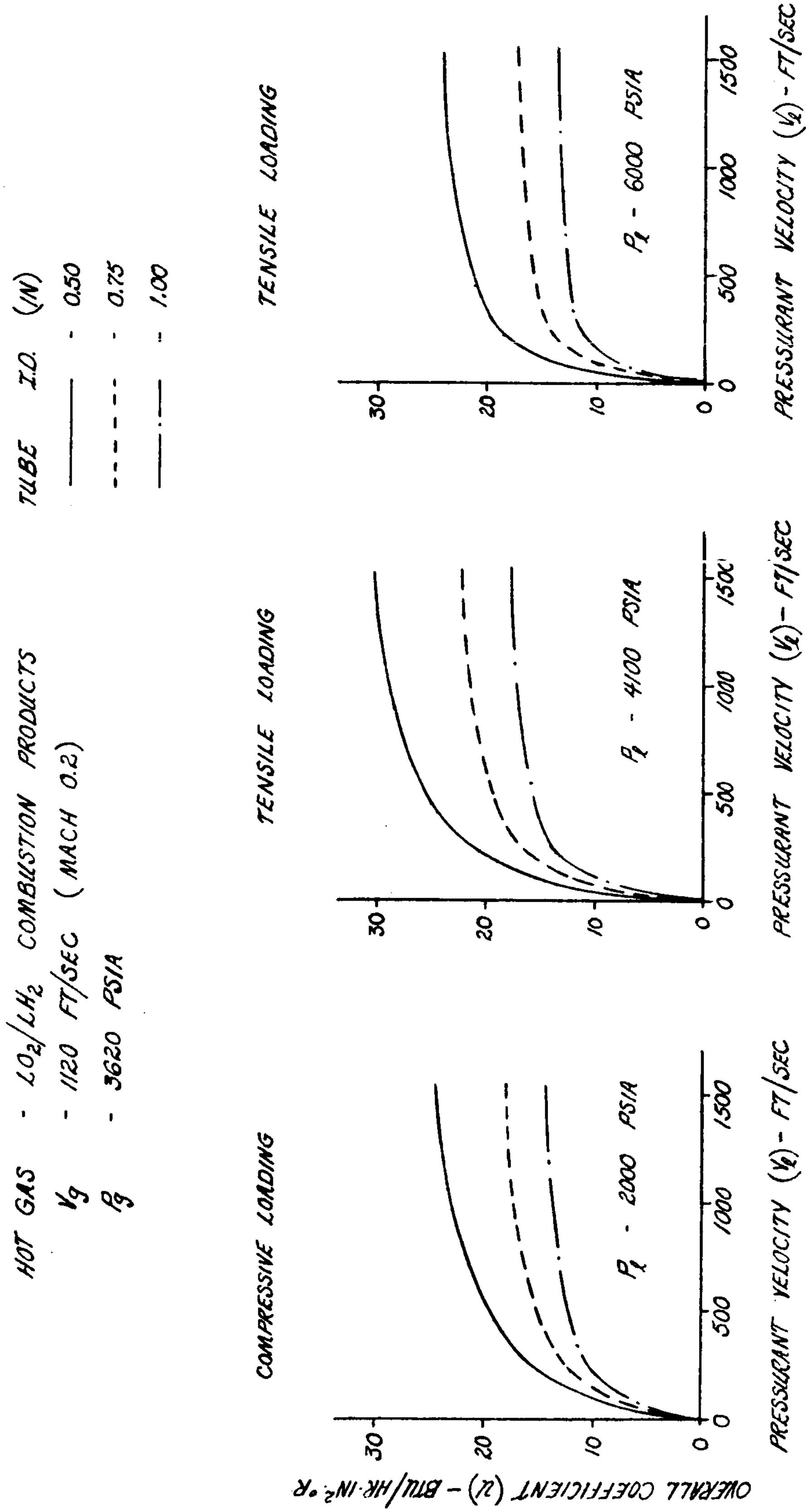
Overall Heat Transfer Coefficient vs Pressurant (LH_2) Pressure

Figure IX-B-9



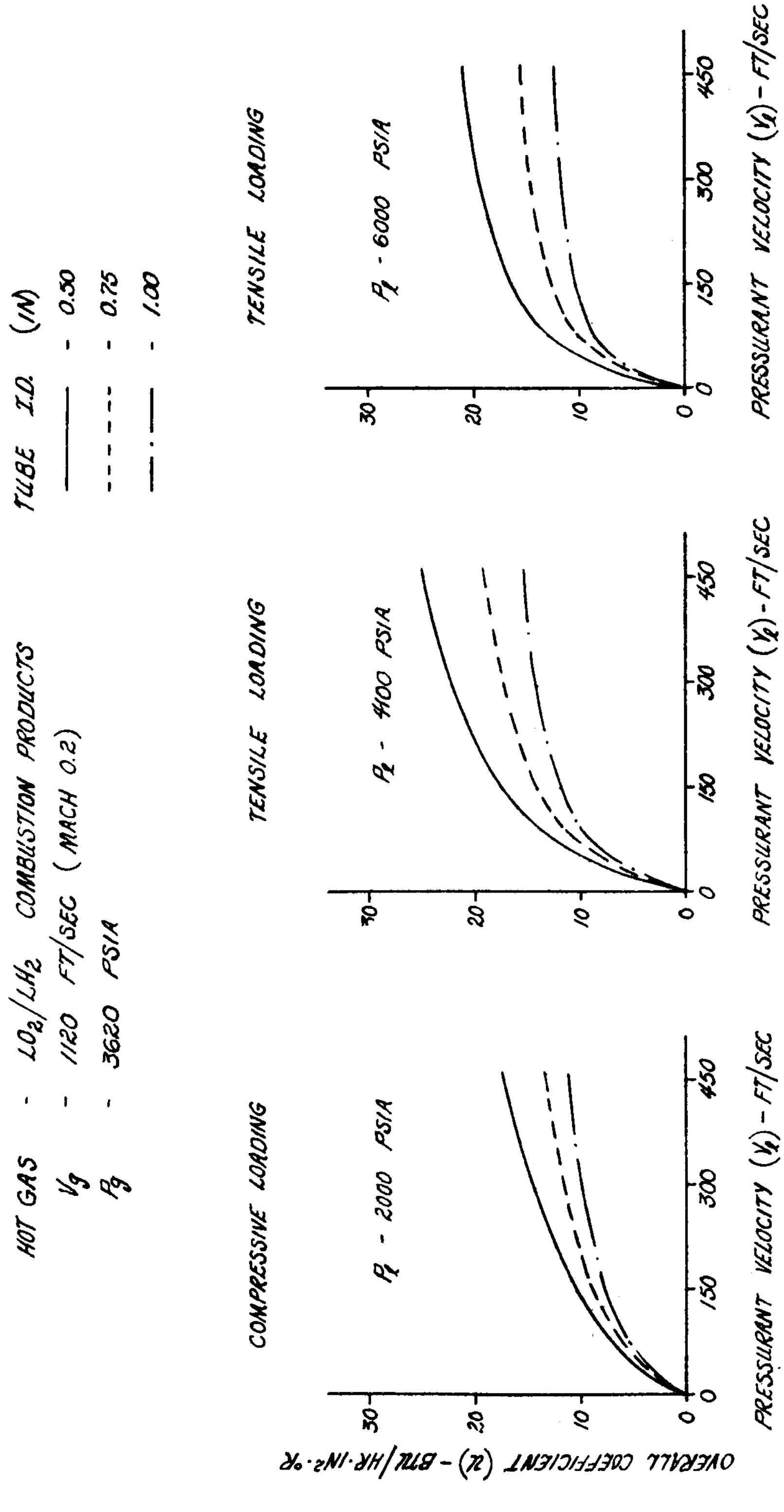
Overall Heat Transfer Coefficient vs Pressurant (LO_2) Pressure

Figure IX-B-10



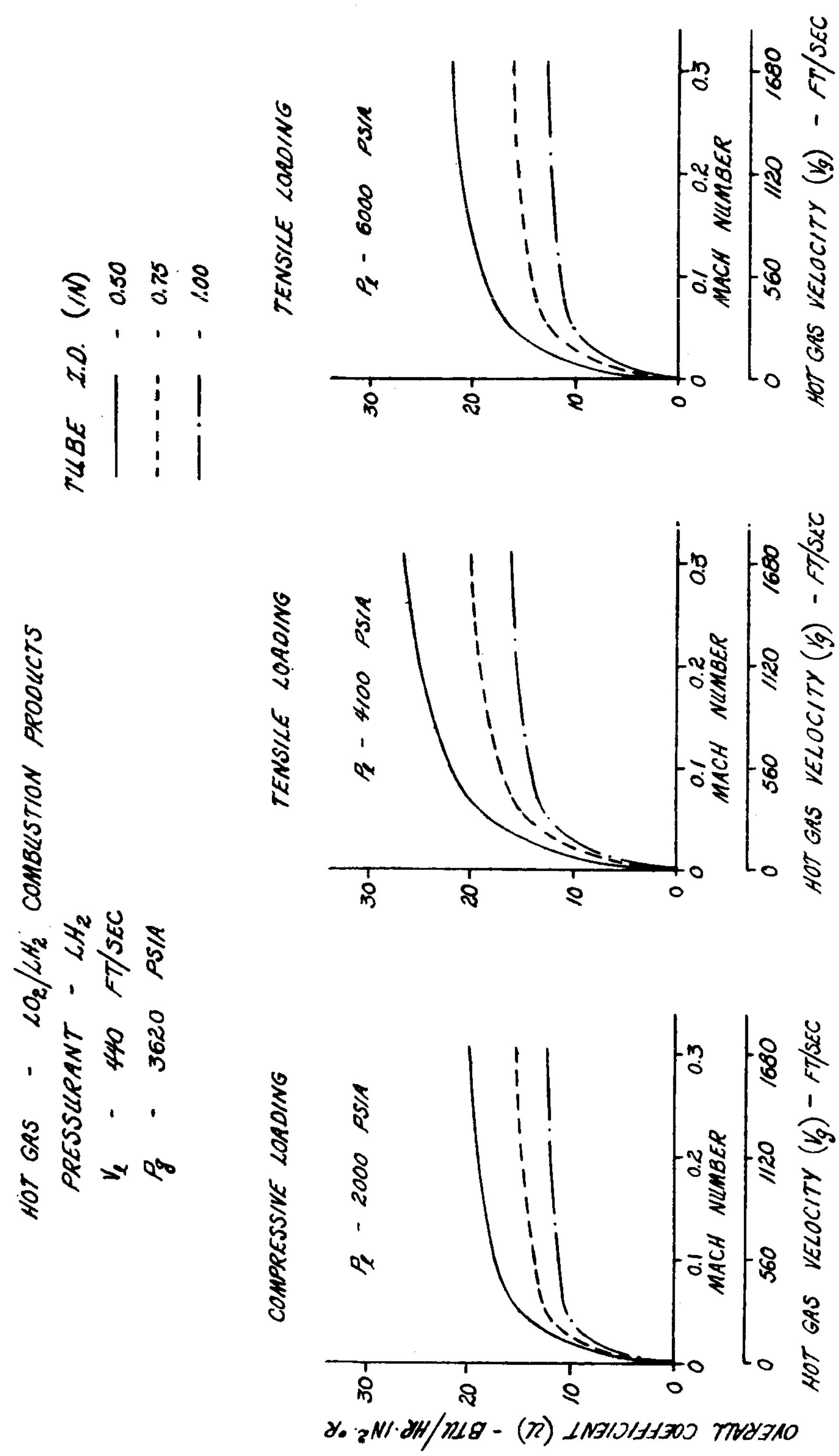
Overall Heat Transfer Coefficient vs Pressurant (LH_2) Velocity

Figure IX-B-11



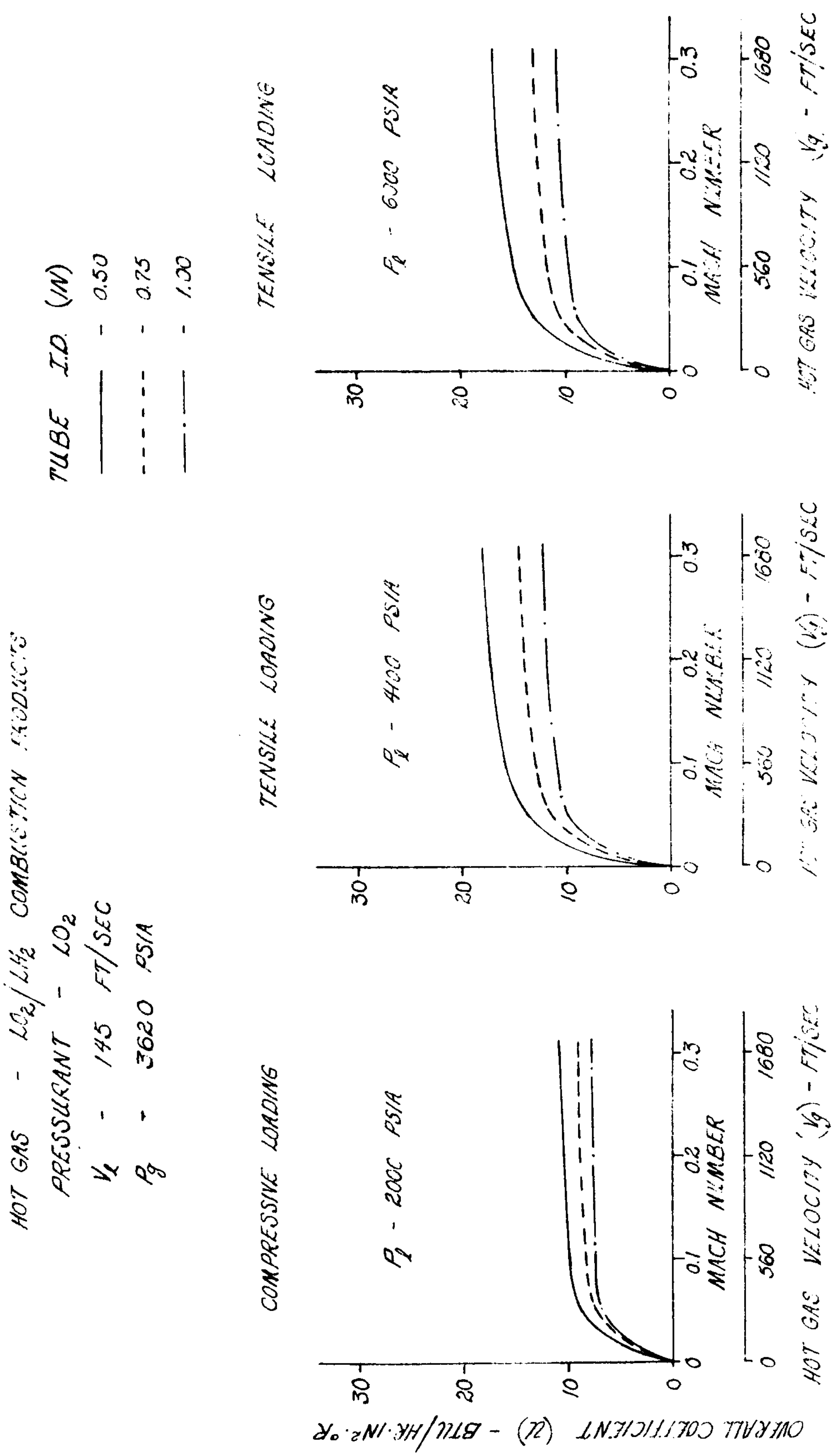
Overall Heat Transfer Coefficient vs Pressurant (LO_2) Velocity

Figure IX-B-12



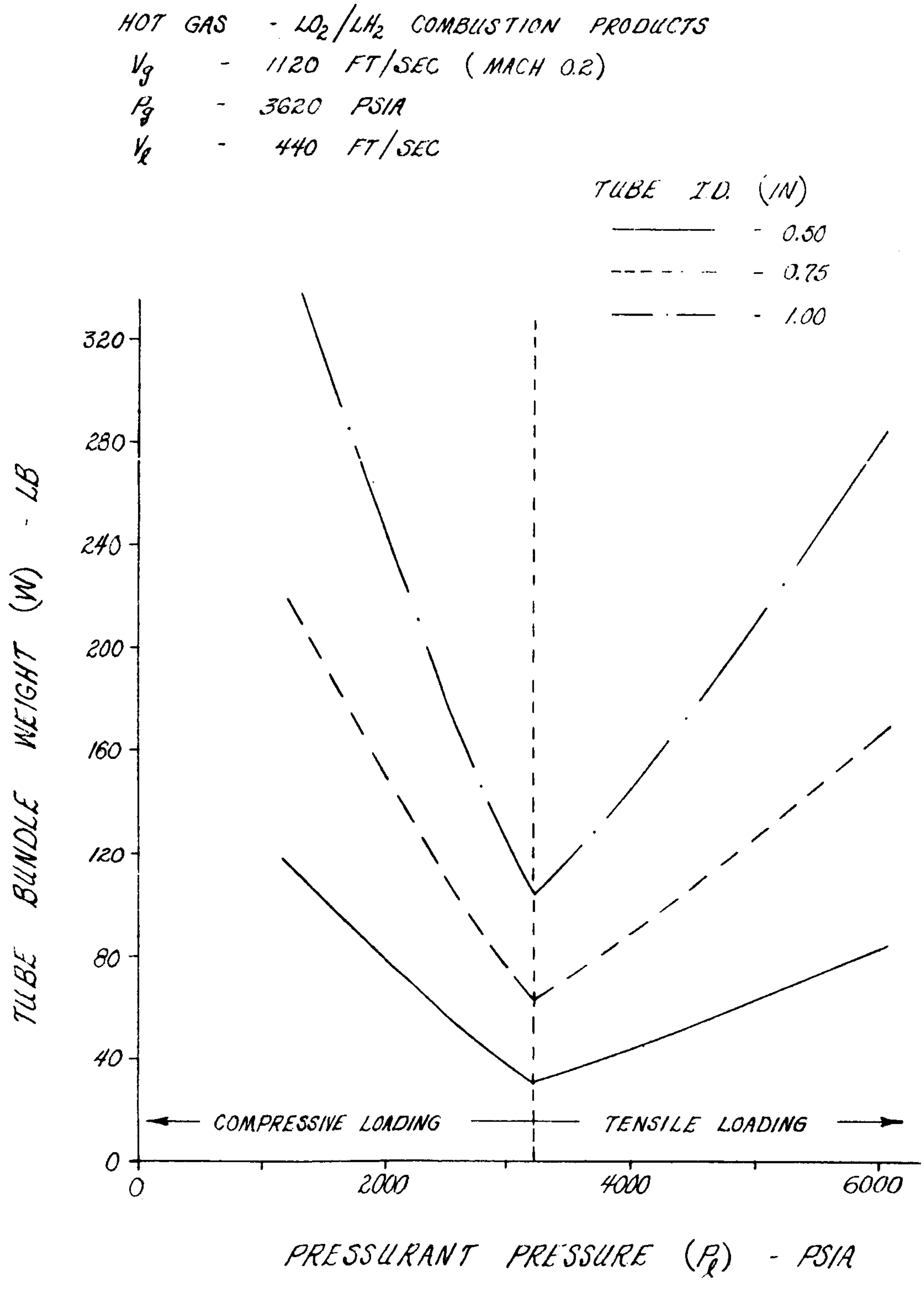
Overall Heat Transfer Coefficient vs Hot Gas Velocity

Figure IX-B-13



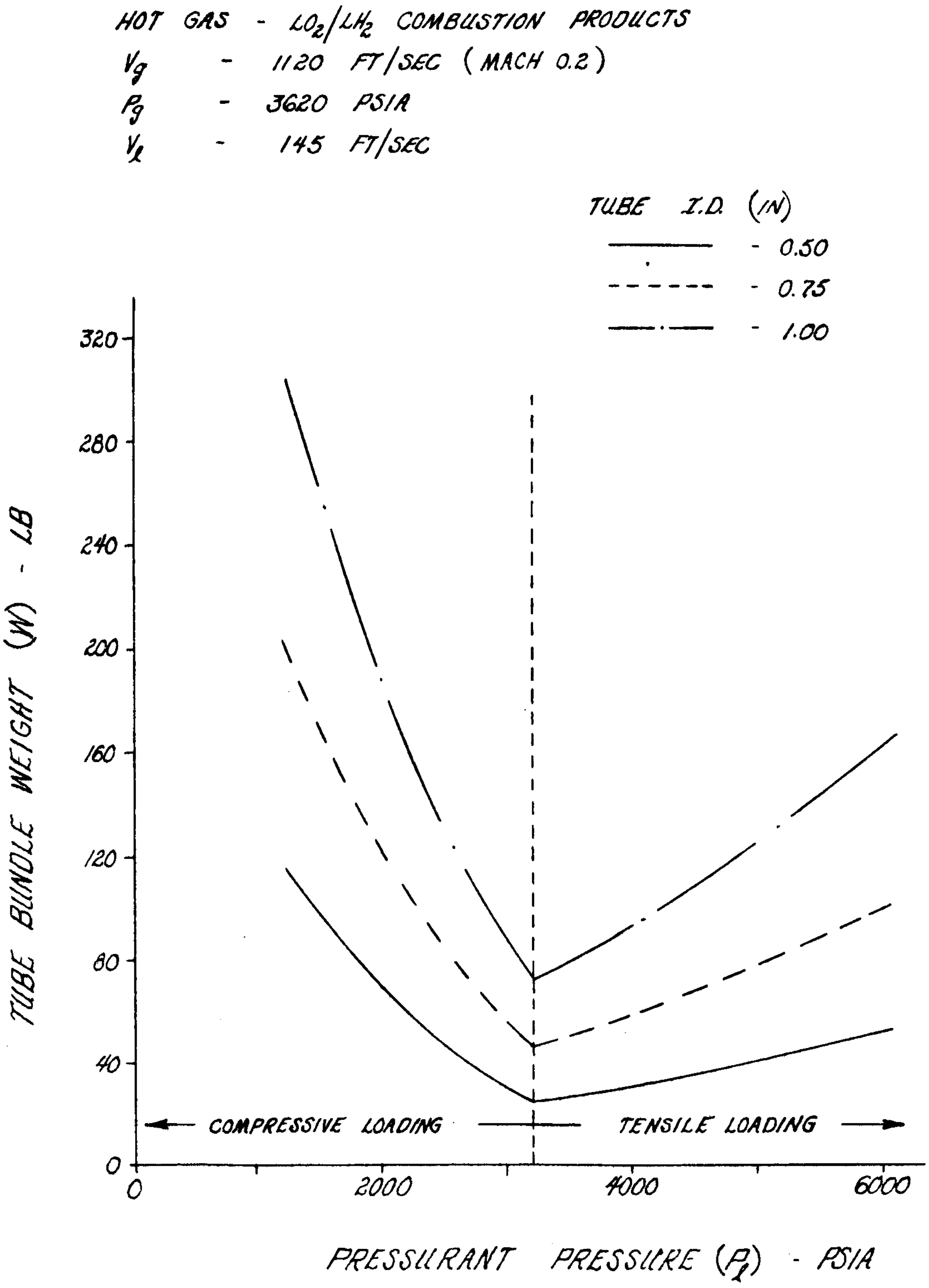
Overall Heat Transfer Coefficient vs Hot Gas Velocity

Figure IX-B-14



Tube Bundle Weight vs Pressurant (LH_2) Pressure

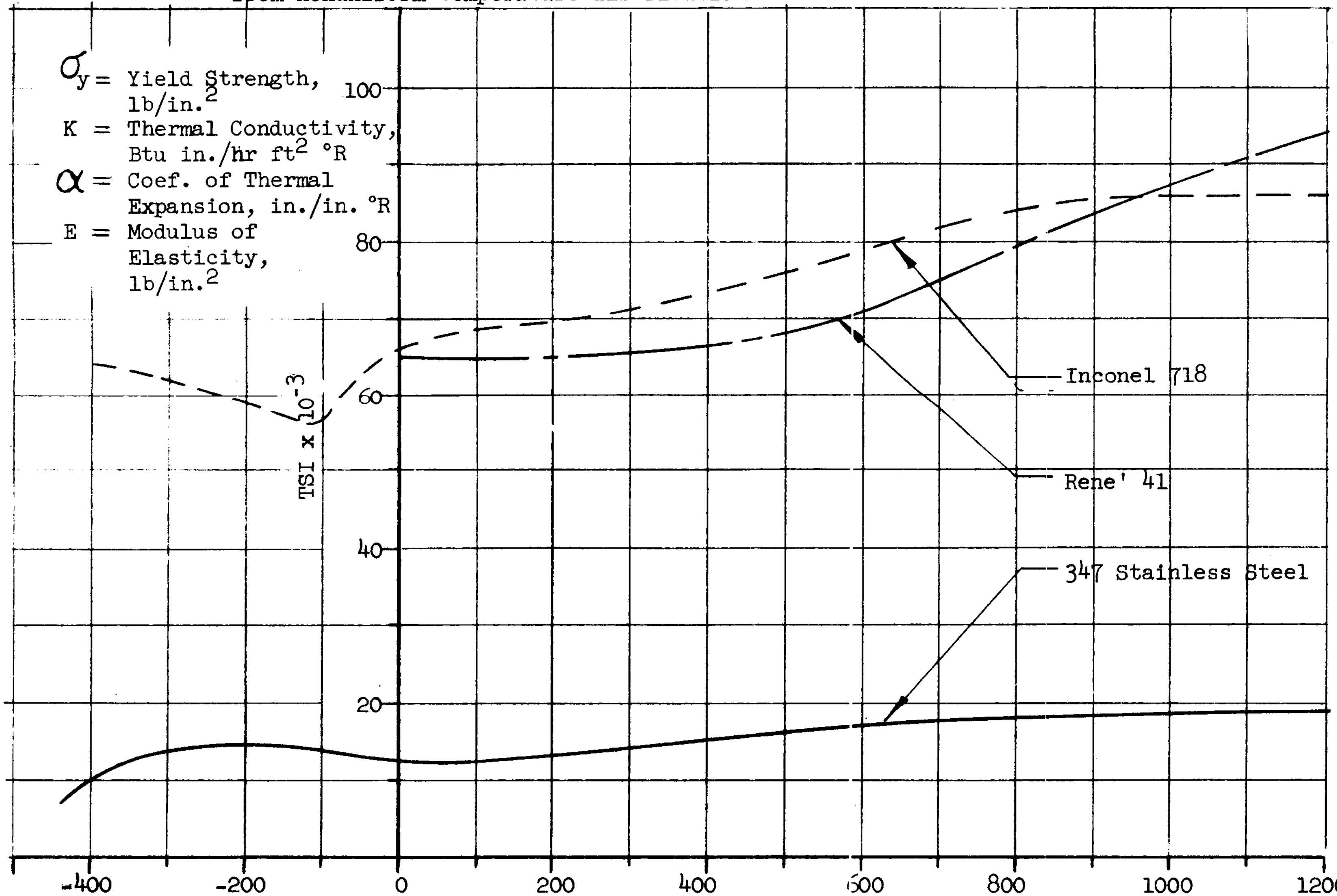
Figure IX-B-15



Tube Bundle Weight vs Pressurant (LO_2) Pressure

Figure IX-B-16

* $TSI = \frac{\sigma_y K}{12 \alpha E}$, a relative indication of a material's ability to withstand thermal stresses resulting from nonuniform temperature distribution.



Thermal Stress Index (TSI) vs Temperature

Figure IX-B-17

IX, Heat Exchangers (cont.)

C. HEAT EXCHANGER DESIGN FOR THE AJ-1 ENGINE

A point design heat exchanger for use on the AJ-1 engine is shown in Appendix H. The design analysis represents a specific application of the methods which were examined parametrically in the preceding section. In addition to the nonvariable parameters selected for the parametric analysis, the fluid bulk pressures (P_l = 4100 psia and P_g = 3620 psia) were selected. The analysis also considered the use of a 0.50 in. inside diameter tube, although for the optimized design various tube diameters must be considered.

The interrelationships between the many factors affecting the design required a series of iterative steps to establish a complete set of design data. These relationships were examined through the use of curves relating fluid velocities and pressure losses, tube numbers and lengths, and surface area requirements. After establishing certain restrictions that provide design simplicity and ease of fabrication, a trial-and-error procedure was followed to obtain the heat exchanger design. The restrictions provided very little flexibility within the design procedure. However, it was not necessary to relax any of these restrictions in order to satisfy all design considerations.

The total weight of the heat exchanger consisting of tube bundle and shell weights was approximately 500 lb. Since the design located the heat exchanger within the engine hardware (turbine exhaust ducts), the shell weight, which would total approximately 400 lb, does not result in proportionately increased engine weight. However, if the design necessitated the use of an external duct, the shell weight must be considered separately. In addition, such associated hardware as support structure, header manifolds, flanges, etc. will add to the overall weight. A typical "wrap-up" factor of 75% applied to the combined tube bundle and shell weights results in a total heat exchanger weight of 875 lb.

IX, Heat Exchangers (cont.)

D. CONCLUSIONS FROM HEAT EXCHANGER STUDY

A review of the parametric and design analyses performed in this study provides the following conclusions on heat exchanger design for large-thrust, high-pressure liquid rocket engines.

1. Heat transfer surface area requirements and associated weights present no technical problems.

2. More emphasis should be placed on ease of fabrication, reliability, and cost than on weight of the heat exchanger unit.

3. The effects of the various design variables are interrelated. Hence, they require a series of iterative steps to establish an optimized design.

4. Tube wall thickness is a major consideration in maximizing heat transfer per unit weight.

5. The same velocity limitations used for smaller heat exchanger designs still apply.

6. The smaller tube diameters result in the lowest heat exchanger weight. This weight reduction resulting from the thinner walls achieved with small diameters is more significant at high pressures than at low pressures.